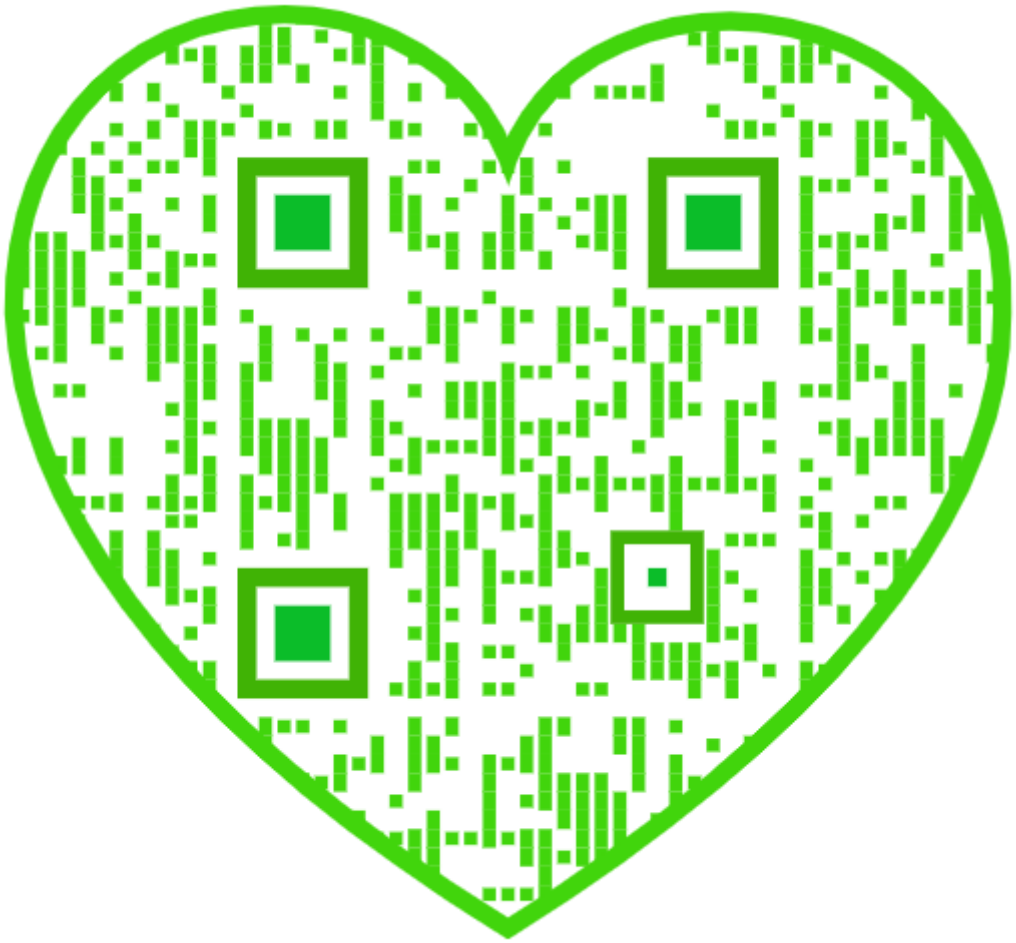
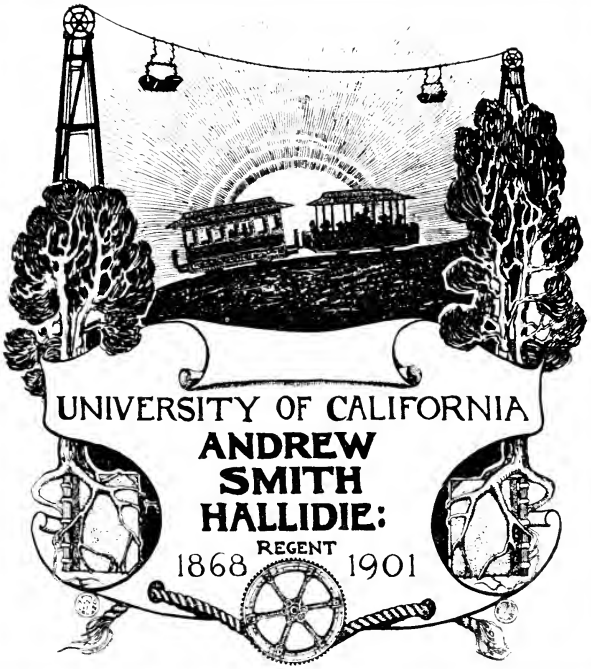



UNIVERSITY OF CALIFORNIA
ANDREW
SMITH
HALLIDIE:
REGENT
1868 1901

THE SLIDE-VALVE AND ITS FUNCTIONS

WITH

SPECIAL REFERENCE TO MODERN PRACTICE

IN THE

UNITED STATES

WITH 90 DIAGRAMS AND ILLUSTRATIONS

BY

JULIUS BEGTRUP, M. E.

HALLIDIE LIBRARY OF THE UNIVERSITY OF CALIFORNIA

NEW YORK
D. VAN NOSTRAND COMPANY
LONDON
E. & F. N. SPON, LIMITED., 125 STRAND
1902

TJ545
B4

HALLIDIE

COPYRIGHT, 1902,
BY
D. VAN NOSTRAND COMPANY.

TYPOGRAPHY BY C. J. PETERS & SON,
BOSTON, MASS., U. S. A.

PREFACE.

THE Slide-Valve has been called the heart of the steam engine, and the simile is not badly chosen; for the valve is a distinct and vital part of the engine, controlling and regulating the circulation of its life-fluid in a manner not entirely unlike that of a living heart. It is the office of the slide-valve to direct the motion and action of this subtile and expensive fluid to best advantage and without waste, so as to make the engine an effective and economical motor.

As the valve must be designed so as to effect an economical steam distribution in the cylinder, it has attained a peculiar significance in scientific steam engine construction; but durability and permanency of form are requirements not less imperative, and they involve constructive problems of a different order. In recognition of this fact an attempt has been made in this work to treat the subject with due regard to the various requirements of modern practice. The fundamental principles are fully explained, and are illustrated by new graphical methods, and a number of special valve constructions are described and analyzed, in order to exhibit in a comprehensive manner how the exacting conditions of higher steam pressure and higher speed have been met by modern engine-builders.

An endeavor has been made to present the subject-matter of this book in a condensed form, as being best adapted to the requirements of practical men, and the author has in this respect followed the suggestions of an extensive personal experience.

The information is presented in more or less explanatory form, which is necessary in order to preserve its scope and general character by the absence of a multiplicity of details; but lengthy explanations are studiously avoided, for it is the author's opinion that practical knowledge — which is the more complete knowledge — cannot be imparted by the use of many words. Nothing is fully comprehended before the learner can follow the thoughts of the teacher, but he must be allowed to learn this in his own way, or by his own efforts, if the acquired knowledge is to be of any actual use. All the book really can accomplish is to start the reader thinking in the right direction, which often may be done by a few carefully selected words.

Both verbal and graphical demonstrations are used, that one form may supplement the other. The verbal treatment is the broader, but the graphical representation is indispensable as an illustrative and explanatory supplement, and as far as the valve motion is concerned, it is the only method practiced by those who build engines, or make the drawings from which they are built. Further, the graphical representation has the advantage that it presents a number of associated facts in one frame, as it were; and it may, therefore, eventually lead to those broad conceptions which are of so great practical utility, and which the best verbal exposition sometimes may fail to disclose.

The valve-diagrams presented in this book have been used for the last ten years, on many different occasions, and they have given more general satisfaction than others which are better known; and the reader will doubtless share this opinion, if the methods here used are accorded a fair trial.

Comparatively few letters of reference are used in the text, and it is believed that this will make it easier to follow the demonstrations.

J. BEGTRUP.

JERSEY CITY, N. J., January, 1902.

TABLE OF CONTENTS.

CHAPTER I.

THE COMMON SLIDE-VALVE.

	Page
Introductory Remarks	1
The Common D-Valve	2
Simplest Form of Slide-Valve	4
The Valve-Motion in Relation to the Piston-Motion	9
Valves Without Lap	11
The Primitive Valve-Diagram	12
Sweet's Valve-Diagram	14
Variable Valve Travel	15
Limitations of the Combination Valve	16
Steam-Ports	18
Port-Opening	19
Valve Dimensions	20
Valve Diagram for Shifting-Eccentric Engines	21
Variation in Port-Opening	24
Notes About Lead	25
Compression	27
The Motion of Locomotive Valves	28
Multiporting	31
Setting the Engine on Dead Centers	32

CHAPTER II.

IMPROVED SLIDE-VALVES.

The Double-Ported Marine Slide-Valve	34
The Allen Locomotive Valve	35
The Straight-Line Balanced Valve	36
Variations of the Straight-Line Valve	38
The McEwen Valve	39
The Ball Telescopic Valve	41
Balancing a Common D-Valve	42
The Richardson Balanced Valve	44

PAGE

THE THOMAS BALANCE 44
PISTON-VALVES 46
VALVES OF THE "IDEAL ENGINE" 47
THE WESTINGHOUSE STANDARD VALVE 49
VALVES ON COMPOUND ENGINES 50
THE WESTINGHOUSE COMPOUND VALVE 53
THE VAUCLAIN VALVE 54
THE ALLFREE VALVE-GEAR 57

CHAPTER III.

FOUR-VALVE SYSTEMS.

INTRODUCTORY REMARKS 61
CORLISS VALVES 62
LIMITATIONS OF THE CORLISS GEAR 65
THE SINGLE-ECCENTRIC VALVE DIAGRAM 67
TWO ECCENTRICS 69
CORLISS VALVE DIMENSIONS 71
DIRECTIONS FOR SETTING THE VALVE-GEAR 72
VALVES OF THE PORTER-ALLEN ENGINE 72
GRIDIRON VALVES 75
THE HILL VALVES 76
THE WHEELOCK VALVES 78

CHAPTER IV.

INDEPENDENT CUT-OFF.

INTRODUCTORY REMARKS 80
THE CUT-OFF VALVE ON A STATIONARY VALVE-SEAT 81
THE CUT-OFF VALVE ON THE BACK OF THE MAIN VALVE 83
FUNDAMENTAL PRINCIPLES 83
THE MEYER CUT-OFF 85
THE MEYER CUT-OFF DIAGRAM 86
LIMITATIONS OF THE MEYER CUT-OFF 89
THE RIDER CUT-OFF VALVE 89
CUT-OFF VARIED BY ROTATING ECCENTRIC ON SHAFT 90
GRIDIRON VALVES WITH INDEPENDENT CUT-OFF 96
BEGTRUP'S ECCENTRIC 97
CUT-OFF VALVE WITH CONSTANT TRAVEL ON MAIN VALVE 99
THE BUCKEYE VALVE-GEAR 102
PISTON-VALVES WITH INDEPENDENT CUT-OFF 105
CUT-OFF VALVE WORKED BY MEANS OF A LINK 106
INDEPENDENT CUT-OFF ON FOUR-VALVE ENGINES 108

PAGE
THE McINTOSH & SEYMOUR VALVES 108
VALVES OF THE RUSSELL ENGINES 110
THE BUCKEYE VIBRATING CUT-OFF 111

CHAPTER V.

THE SLIDE-VALVE ON PUMPS.

BLAKE 117
DEAN BROTHERS 119
KNOWLES 121
DAVIDSON 121
CAMERON 123
WORTHINGTON 124

CHAPTER VI.

ANGULARITY OF CONNECTING-ROD AND ECCENTRIC-ROD.

THE ANGULAR MOTION OF THE CONNECTING-ROD 128
THE ANGULAR MOTION OF THE ECCENTRIC-ROD 132
STEAM- AND EXHAUST-LAPS 134
EQUALIZING BOTH LEAD AND CUT-OFF 134
THE INCLINED ROCKER FOR SHIFTING-ECCENTRIC ENGINES 136
EQUALIZING THE LEAD ON SHIFTING-ECCENTRIC ENGINES 137
UNEQUAL LAPS ON VARIABLE CUT-OFF ENGINES 139

HALLIDIE LIBRARY
OF THE
UNIVERSITY
OF
CALIFORNIA

THE SLIDE-VALVE AND ITS FUNCTIONS.

CHAPTER I.

THE COMMON SLIDE-VALVE.

INTRODUCTORY REMARKS.

THE term "**slide-valve**" is applied to flat-faced valves having a continuous reciprocating sliding motion, whereby steam is alternately admitted to and exhausted from a cylinder; and this term will also apply to similar parts of gas, air, and water engines and pumps. The steam engine slide-valve and some typical pump valves will be considered here.

The sliding surfaces may be cylindrical, and this does not affect the principles of the valve motion; but in such case different names are used, as **piston-valve, Corliss-valve, semi-rotary or oscillating valve,** etc. These valves are only modifications of the original slide-valve, and they will be included as such here.

Slide-valves, of whatever design, can easily be referred to and compared with the common D-valve, their functional properties, in all cases, being closely related. But the **D-valve** — so called because in section slightly similar to the letter D — is not the simplest slide-valve, *per se;* and in order to fully understand the principles and master the analysis of the subject, it is advisable to commence with a careful study of the simplest elementary form.

By thus taking up the elements of the subject, or going to the bottom of it, as it were, the consequent study of more or

less elaborate forms and combinations will be greatly facilitated; and such a course may lead to broad and independent views, so essential to successful and progressive engineering.

The diagrams presented in this chapter are not supposed to represent actual practice. They are only here to illustrate and explain the text, and to give **graphical directions** for laying out the essential parts of valve and gear, and no attempt is made to show details. Some of the sketches are purposely out of proportion, for otherwise almost unlimited space would be required, or else clearness and distinctness of the principal parts would be sacrificed; and it can hardly be considered a serious defect, for some imagination is a prerequisite in constructive engineering, for which there is no substitute.

THE COMMON D-VALVE.

Fig. 1 shows the valve in its central position and part of the cylinder. By moving the valve to the right steam is exhausted from the right hand side of the cylinder and admitted to the left hand side; and by moving it to the left steam is exhausted from the left hand side, and admitted to the right hand side. Steam is admitted past the outer edges of the valve, and the exhaust escapes past the inner edges into the exhaust-cavity, and from thence to the exhaust-pipe (not shown).

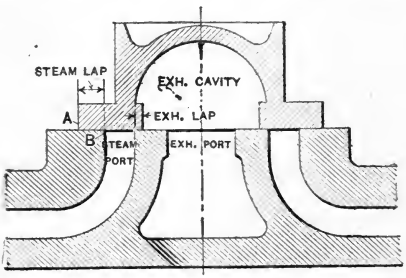


Fig. 1.

The flat surface on which the valve slides is called **the valve face of the cylinder** or the **valve-seat,** and the corresponding surface of the valve is the **face of the valve.** The orifices of steam-passages, covered by the face of the valve, are the **steam-**

ports, and the central opening in the cylinder casting is called the **exhaust-port.**

The conventional use of this term is somewhat unfortunate, for, as the exhaust from the cylinder passes out through the same ports through which steam is admitted, these ports are also entitled to be named exhaust-ports. True, the terms steam and exhaust signify here only different conditions of the steam, but the name "exhaust-port" for the middle opening conveys unconsciously the impression that the two exterior ports are for steam only, and the central port is for the exhaust. Besides, the name port is too dignified in this connection, where it only means a hole or opening through which the exhaust escapes **after it has left the steam-port,** and no particular importance attaches to this hole; it can be of any shape and size, provided it is big enough; in fact, it hardly needs a special name.

The steam-ports, on the other hand, are essential and characteristic components of the valve-face, for in conjunction with the valve, they regulate the steam distribution in the cylinder, and their location and dimensions require careful consideration. Engines having separate exhaust-valves have also separate exhaust-ports, but these bear no relation to the "exhaust-port," so called, of the common D-valve. The force of this argument will become apparent later on.

That part of the face of the valve which overlaps the outside edge of the steam-port, when the valve, Fig. 1, is in its central position, is called **steam-lap,** and the over-lapping of the inside edges is called **exhaust-lap.** The terms "inside" and "outside" lap are often used, but are not to be recommended, for many valves have steam-lap on the inside and exhaust-lap on the outside. Valves are often constructed without exhaust-lap, and occasionally the exhaust-edge of the valve does not reach to the port-edge, in which case the face of the valve does not entirely cover the ports in its central position, and the uncovered part is then referred to as **negative exhaust-lap** or

exhaust clearance. If not otherwise specified, lap means steam-lap. The two partitions which separate the steam-ports from the exhaust-opening are named **bridges.**

It is proper to study the slide-valve as it relates to the action of steam on one side of the piston only; for in a double-acting cylinder, being, as it were, a combination of two single-acting cylinders, the action on one side of the piston is practically an exact repetition of that on the other side; and the D-valve can be divided into two equal halves, each of which will take care of the steam on one side of the piston; and in order to place the back pressure in its proper relation to the forward stroke, it is only necessary to consider the exhaust-action in reversed order.

The duty of the slide-valve, broadly speaking, is to admit steam during the forward stroke and exhaust it during the return stroke, and which includes expansion and compression.

The majority of engines are double-acting; that is, steam is admitted alternately on both sides of the piston, and the common slide-valve is charged with the fourfold duty of admitting and releasing steam at both ends of the cylinder; and we may conceive this being attained by a unification of four valves in one body. The regularly alternating periods of admission and exhaust, and the symmetrical arrangement of steam-ports admit of a single valve being used.

SIMPLEST FORM OF SLIDE-VALVE.

Fig. 2 represents a slide-valve, whose only duty is to open and close for admission of steam to **one side of the piston.** When it is moved to the right, a sufficient distance, it uncovers part of the port and admits steam to the cylinder, and by its retrograde motion the port is closed again. It controls the admission of steam only, and there will have to be another valve to open and close for the exhaust; and in case of a double-acting engine two such pairs, or four valves in all, would be needed.

The reciprocating motion of slide-valves is generally produced by eccentrics attached to the crank shaft of the engine, and an eccentric — being nothing but an enlarged crank-pin —

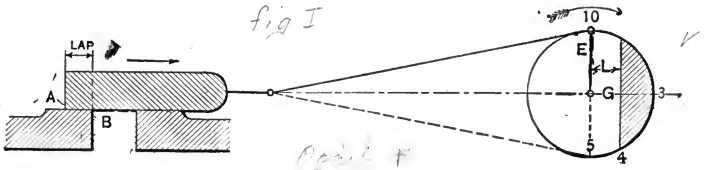


Fig. 2.

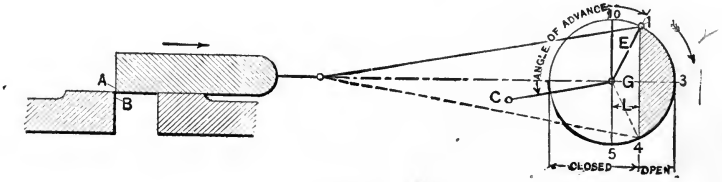


Fig. 3.

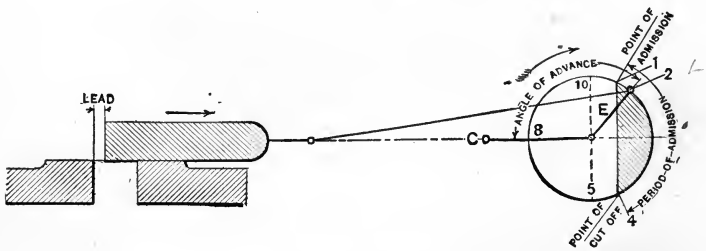


Fig. 4.

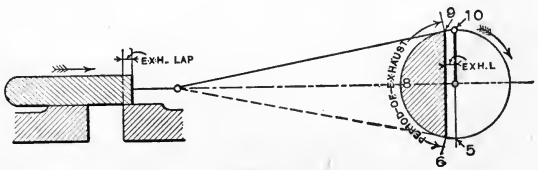


Fig. 5.

is in the annexed diagrams represented by a small crank, or arm, E, and which communicates motion to the valve through a rod, as indicated. The diameter of the circle described by

the center of the eccentric is called **the throw of the eccentric,** and it represents here the full stroke or **travel of the valve.**

According to this definition the throw of an eccentric is twice its eccentricity or radius; but, as the throw of a crank means the distance from center of shaft to center of crank-pin, it ought to mean the same when speaking of an eccentric. However, rhetoric is not cultivated in the shop or engine room, and when they talk about the throw of an eccentric it is well understood what it means.

For convenient representation in a limited space, the distance between eccentric and valve is made short in the diagram, which makes the angularity of the eccentric-rod appear abnormal. In ordinary stationary engines this angularity is always very slight, and its effect on the valve motion may be ignored; and for this reason we may assume the eccentric-rod parallel with its medial-line of action, as also would be the case if we assume an eccentric-rod of infinite length.*

Fig. 3 shows the valve in the position where it commences to open the steam-port. Edge A of the valve and edge B of the port are now "line and line," and by continued turning of the eccentric in the direction shown by the arrow, these edges will separate, leaving an increasing opening for the passage of steam, till the eccentric has reached the extreme of its throw, when by further turning it will push the valve back towards edge B; and when the eccentric is in position 4, vertically opposite position 1, the port is closed again.

It should be observed that we are here only considering **one edge of the valve and one edge of the steam-port;** other relations of port and valve being for the present quite immaterial. These edges come together twice during each revolution of the eccentric, when it is at the vertically opposite points 1 and 4,

* The actual effect produced by the angular motion of the eccentric-rod is explained in the last chapter of the book.

and the valve, therefore, opens and closes the port **once during each revolution.**

If arc 1–4 represents the period during which the port is open, arc 4–1 will represent the period during which it is closed. The greatest port opening is equal to the distance G-3, and for any position of the eccentric, the corresponding port opening can be measured horizontally on the shaded segment. When the eccentric is at **"half throw,"** at point 10, the valve is in the position shown in Fig. 2, and the amount it overreaches the port in this position is the **"lap,"** and which is equal to the distance L. Thus **the length of the period during which steam is admitted depends on the amount of lap,** for the valve must move from its middle position a distance equal to the lap before it uncovers the steam-port, and it is evident, that the longer the period during which the port is closed the shorter must be the period during which it is open. A correct idea of this relationship can be had by an inspection of the figure, which clearly shows how an increase or diminution of distance L affects the period of steam admission.

Comparing the valve here described with that shown in Fig. 1 it will be seen, that by cutting off one-half of the latter and filling up the exhaust-cavity, we have a valve very similar to that shown in Fig. 2, and operating in exactly the same manner. It will also readily be seen that a valve like that shown in Fig. 2, working in a separate chamber, could be made to open and close an exhaust-port in much the same manner as the steam-valve controls the admission of steam or as one of the inner edges of the D-valve, Fig. 1, controls the exhaust, and that the period of exhaust for each turn of the eccentric would depend on an exhaust-lap corresponding to the steam-lap in Fig 2 or the exhaust-lap in Fig 1.

Fig. 5 shows an exhaust-valve and its eccentric at half throw; the main difference between this valve and the steam-valve, Fig. 2, being that in Fig. 5 the valve is turned end for

end and has a smaller lap, which makes the period of exhaust longer than the period of admission. It would also be necessary to provide means for holding this valve to its seat, which is not shown.

The period of exhaust is represented by the arc of the shaded segment, Fig. 5. If the exhaust eccentric is fixed on the shaft in the same position as the steam eccentric, the period of exhaust will, in a manner, be diametrically opposite the period of admission; that is, if we imagine the period of revolution represented by a circle. Under such conditions, it is evident that all four valves for a double acting engine might be driven by one eccentric; and by uniting the four valves in one body, and by providing an exhaust cavity to keep the exhaust separate from the live steam, we arrive at the construction shown in Fig. 1.

It should be noted again, that here the exhaust passes through a port which also serves for admission of steam, and that steam passes over the outer edge of this port on its way to the cylinder, and is exhausted from the cylinder over the opposite edge, and that the steam- and exhaust-lap, therefore, extend in opposite directions, as on the separate valves Figs. 2. and 5. Each operation of this valve, therefore, depends on the relative position of two individual edges, just as does each operation of the simple valves referred to; and thus is established the proposition that such simple valves are, in a sense, integral parts of the D-valve, or, in other words, **the D-valve is equivalent to four simple valves driven by one eccentric.**

Let the circle in Fig. 6, p. 12, represent the path of the center of the eccentric, and arcs 1–4 and 6–9 represent steam and exhaust periods respectively, during one revolution of the eccentric, for the left half of the D-valve, and the left side of the piston. It is here observable that the greatest steam and exhaust openings occur when the valve is at the two extremes of its travel, that is, at **diametrically opposite points of the**

period of revolution; that any opening of the port can be measured horizontally on the segments; that if there be no exhaust lap, the exhaust period will cover one-half revolution; and that **lap plus port-opening equals half travel of valve.**

THE VALVE–MOTION IN RELATION TO THE PISTON–MOTION.

The main object of the discussion so far has been to establish a clear conception of the simple relationship which exists between laps and periods of admission and exhaust. The next subject for consideration is the relation of these periods to the motion of the piston in the cylinder, which includes determination of the position of the eccentric relative to the crank. It has been shown how these periods may be lengthened or shortened by changing the laps of the valve; and any one of them can evidently be made to coincide with any period of the piston-stroke by fixing the eccentric in a suitable position on the crank-shaft.

First: Where should the valve be when the piston is at the end of the cylinder and commencing its forward stroke? As the port must open at or near the beginning of the piston-stroke, when the crank is on its first **"dead center,"** and as the period of admission occurs during the opposite half-revolution of the eccentric, and must be considerably less than one-half period of revolution, it follows that the eccentric must be past the middle of its throw, or **more than one-quarter of a revolution ahead of the crank.** Let C, Fig. 3, represent the crank when the valve is at the point of admitting steam, at 1, then considering that eccentric and crank are moving together with the crank shaft, it is clear that when the crank has arrived at the end of its return stroke, at C, Fig. 4, the eccentric has advanced the same angular distance, from position 1 to 2, and there will be a small opening between the valve and the port-edge, as shown; and during this short interval steam enters the cylinder **in ad-**

vance of the piston. This initial port opening is called **lead** or sometimes **steam-lead,** in contradistinction from **exhaust-lead,** which means the opening for exhaust at commencement of the return stroke. Steam-lead is usually supposed beneficial, and the desirability of exhaust-lead is unquestionable. **Lead is increased by turning the eccentric forward on the shaft, and is diminished by turning it backward.**

Sometimes, under certain conditions, the port opens after the forward stroke has commenced, and in such case the expression **"negative-lead"** may be used, to indicate that the valve overlaps the steam-port to a certain extent **at the commencement of the stroke.**

Let the position of the eccentric at the beginning of the stroke be at 2, Fig. 4; then period of admission from the time the piston starts on its forward stroke is represented by arc 2–4, which obviously always should be considerably less than the period of one-half revolution in order to provide for expansion of the steam. The angle 10–2 has been named the **angle of advance,** but, thus should be named the angle between the crank arm and eccentric arm; for it represents how much the eccentric actually is in advance of the crank. Any other designation is confusing, and there is no valid reason for any other use of this term. If there ever was a good reason for it, it does not exist now. In this book, **the "angle of advance" means the angular advance of the eccentric relative to the crank,** as represented by angle 8–2, Fig. 4 and 6.

When, by continued turning of the shaft, the eccentric reaches position 4, the admission of steam to the cylinder is cut off. Where will the piston then be? Mark off the angle of advance from point 4 in a reverse direction, and a radial line intersecting the crank-circle marks the location of the crank-pin, and its horizontal projection will indicate approximately the position of the piston in the cylinder when the admission of steam is cut off and expansion commences. The exact location

of this point — **the point of cut-off** — is sometimes required, and the angularity of the connecting-rod must then be taken into account. (See last chapter.)

As steam always is admitted at a point very near the commencement of the stroke, the period of admission must be made shorter or longer according as an earlier or later "cut-off" is required. This can be effected by lengthening or shortening the steam-lap; but as the point of admission is fixed near the commencement of the stroke, the period **as a whole** must occur earlier or later according to the location of the point of cut-off, and the angle of advance must be changed accordingly — which will become quite clear by an inspection of Fig. 4. Thus is established the important rule: **Period of admission is shortened by increasing the lap and the angle of advance, and is lengthened by diminishing the lap and angle of advance.** Also mark: that **all periods of steam distribution occur earlier or later in the stroke, according as the eccentric is turned ahead or back on the shaft.**

VALVES WITHOUT LAP.

Valves have been used without steam- and exhaust-lap at a time when the gain by expansion was not fully recognized and compression of exhaust had no significance. When such valves have their eccentrics at right angles to the crank, the points of admission and release of the steam occur when the crank is on its "dead center"; but the eccentric was presumably moved forward some to give the valve-lead. Such valves are not used now, and they may be classed with historical curiosities. In treatises on the steam-engine this lap-less construction is usually discussed at length as an introductory to a chapter on slide-valves, though the discussion of this special and obsolete construction seems hardly profitable, and it is not conducive to a broad, practical conception of the general subject.

THE PRIMITIVE VALVE–DIAGRAM.

Fig. 6 is a combination of diagrams 4 and 5, and it represents a simple D-valve diagram. The crank is supposed to be at its dead center, or at the extremity of its throw, at the commence-

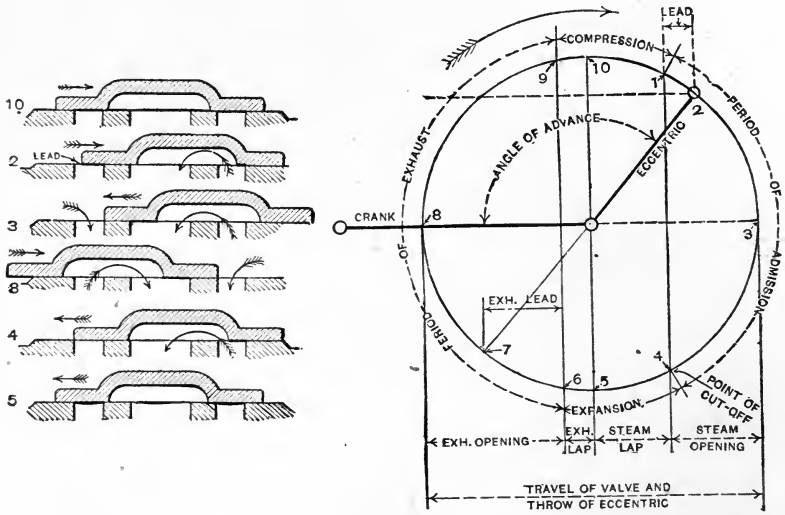


Fig. 6.

ment of the forward stroke, and significant positions of the valve during one revolution of the crank are indicated diagrammatically to the left. The various positions of the eccentric are marked on the eccentric-circle in the following order.

1. Point of admission, just before end of return stroke.
2. Lead at **commencement of forward stroke.**
3. Extreme throw of valve, port wide open.
4. Steam is cut off, expansion commences.
5. Valve in central position.
6. Point of release.
7. Exhaust-lead, at **commencement of return stroke.**
8. Extreme throw of valve, exhaust wide open.
9. Exhaust-closure; compression commences.
10. Valve in central position.

This diagram exhibits in a natural manner all that is required for a full understanding of the functions and principles of construction of a common slide-valve, and from it the position of the valve relatively to the ports, at any point of the stroke, can readily be ascertained by marking off the angle of advance on the eccentric-circle. Note that in this diagram the eccentric-arm is supposed to be moved in unison with the crank-arm, as it does move in reality.

The simple and direct relationship between eccentric and valve motion, as here assumed, is often changed by the intervention of a special valve-gear; but the resolution of such cases presents no special difficulty. Usually the valve-motion is assumed, and the throw and position of the eccentric are then deduced from the design of the valve gear.

It should be particularly noted that the valve-end of the eccentric-rod must be in the same position when the valve opens and closes the port, and that, therefore, each pair of points, 1–4, and 6–9, which mark the beginning and end of the admission and the exhaust period respectively, must be equally distant from a certain point which marks the location of the eccentric-rod pin when the valve and port edges come together, and that, therefore, if the valve-end of the eccentric-rod travels in a straight line, which extended passes through the center of the crank-shaft, chords 1–4 and 6–9 must be parallel. A slight deviation from this direction, caused by the swing of an ordinary rocker arm, is of so little consequence that it may be entirely ignored.*

The foregoing discussion presents the fundamental principles which must govern the construction of slide-valves of any description, and it will serve as an introduction to the study of other forms of valves.

* The general effect of a rocker motion in other directions than the center of the shaft is discussed in the last chapter of the book.

SWEET'S VALVE-DIAGRAM.

In Fig. 6 the various positions of the valve relative to the port during a complete revolution of the eccentric are indicated by the eccentric-arm, as a pointer; but if the circular figure be turned backwards through an angle equal to the angular advance of the eccentric, without disturbing the crank, all the points in the circumference will be placed in the same position relative to

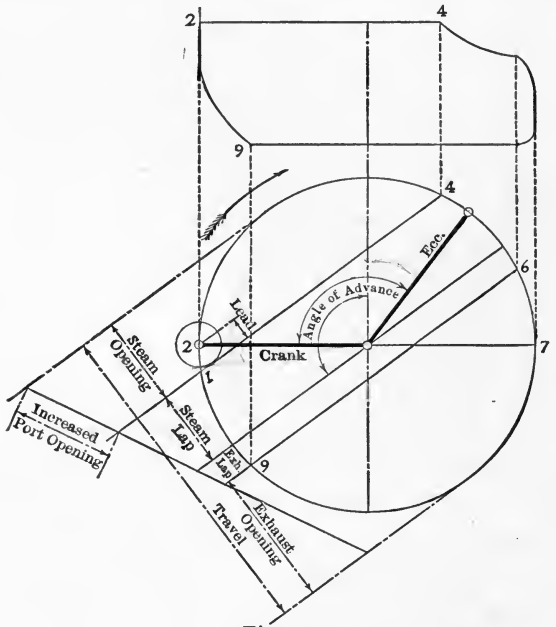


Fig. 7.

the crank-arm as they now occupy relative to the eccentric-arm; and if then the crank-arm, or its center line, be moved into any new position, the corresponding position of the valve will appear directly from the marks in the circumference of the circle. A valve-diagram thus modified is shown in Fig. 7. Note that valve measurements and port-openings are taken at right angles to the oblique lines, and that **these lines must be parallel.** For con-

venience, the eccentric-circle is drawn full size, and the crank-circle is reduced, so as to make both circles appear of same size.

From this circular diagram a theoretical indicator diagram can readily be constructed, as shown, which adds greatly to its usefulness ; for to many engineers the indicator diagram exhibits and explains, at a single glance, all the peculiarities of the steam distribution in the cylinder, and it sometimes contains a whole story in graphic language.

This valve-diagram, slightly modified, was first published by Professor John E. Sweet in the American Machinist of August 30, 1884, and I know no reason why it should not be named the **"Sweet valve-diagram."** No attempt should be made to use the same diagram to indicate different valve proportions or settings, for it makes it confusing to the eye, and it is entirely unnecessary, as half a dozen such diagrams can be drawn in "less than no time."

In order that the opening and closing of the ports may take place at the proper moments during the forward stroke and return stroke respectively, the eccentric must be set at a certain angle in advance of the crank ; and its location depends, therefore, on the **direction** in which the crank turns, though the angle of advance is the same in either case. A reversal of the crank motion would change the valve diagram, only so far as all points above the center line would occupy corresponding positions below it, and *vice versa*, — as if the diagram were turned around its horizontal diameter.

VARIABLE VALVE TRAVEL.

A glance at the diagram shows that **the longer the lap the shorter the period of admission, and the smaller the port opening;** but it should also be noted that the length of periods of admission and exhaust depends absolutely on the **relation between lap and travel of valve,** and that, therefore, shortening the travel has the same effect as increasing the lap, so far as

these periods are concerned; but shorter travel yields smaller openings for admission and release.

It has been shown how the location of the point of cut-off can be changed by changing the lap of the valve and the angular advance of the eccentric, and it is evident that this may also be done by changing the throw of the eccentric and its angular advance, without changing the lap. For instance: by diminishing the throw the period of admission becomes less, and by advancing the eccentric the proper lead is obtained, which will become clear by an inspection of diagrams 6 and 7. Improved slide-valves are often driven by shiftable eccentrics whose throw and angular advance are automatically changed so as to vary the point in the stroke where steam is cut off, as, for instance, on many high-speed engines; or the same result is obtained by hand adjustment, as on the locomotive.

LIMITATIONS OF THE COMBINATION VALVE.

The main limitations of the D valve, or combined steam and exhaust valve, as a steam distributer, is due to the fixed union of its steam and exhaust edges, as when separate steam and exhaust valves are driven by means of one rod. Maximum steam and exhaust opening must occur when the eccentric is at either end of its stroke, just in the middle of periods of admission and exhaust respectively; and there being only one eccentric, this must occur at **diametrically opposite points in the period of revolution.** If, for instance, the admission period is fixed, the exhaust period cannot be arbitrarily changed; for, the eccentric being fixed on the shaft to suit the admission period, we can only vary the exhaust period by varying the exhaust lap, but this will give a variation in both directions; that is, if the exhaust opens earlier it will close later, and if it opens later it will close earlier. The cycle of contiguous events, as marked on the eccentric circle, shows this clearly; also that an early cut-off of the steam in the cylinder is followed by early release

and early exhaust closure, and consequently by much compression of the exhaust. It is therefore quite common practice, when not exhausting into a condenser, to omit the exhaust lap. Moving the exhaust edge back still farther would probably give too early release.

It should be noted that the functional restrictions here mentioned are inherent in any combination of a steam and exhaust valve **conjunctively driven by a single eccentric,** because such valve-gear must derive all its motions from the eccentric, and its alternations must follow closely those of the engine, without reference to the nature of the valve-gear.

The increased compression which follows an early cut-off is sometimes considered an unavoidable evil or inherent defect, and sometimes a desirable feature or incidental advantage of the combination slide-valve, according to existing conditions or various theories.

It should be noted that by an early cut-off the steam opening is much restricted, and particularly so near the point of cut-off, where the steam current is comparatively rapid. The most natural remedy would be an increase of the valve travel, but in case of an unbalanced valve this would increase the unbalanced area and the friction proportionally. When it is desired to limit the valve travel, duplicate or supplementary steam-ports are sometimes resorted to. The restriction of the port-opening, however, is of little consequence, unless the steam is cut off very early in the stroke; and the attendant early release is probably the most objectionable feature. Very early cut-off is only used in connection with a shiftable eccentric or with a link-gear; and the discussion of this case will be resumed later.

The central exhaust cavity of the D-valve removes pressure from the front of the valve, and causes it to be held firmly to its seat by steam pressure on its back; but the unbalanced pressure is always excessive, and in order to reduce it a

pressure-balancing device on the back of the valve is often resorted to.*

On long-stroke engines the steam passages become very long unless the valve is lengthened correspondingly, and the exhaust passage between the valve and cylinder makes it impossible to bring the valve very close to the cylinder-bore ; but these objections are only of consequence when it is desired to make the steam passages short and direct. Great adaptability, simplicity, and compactness are characteristics of the common D-valve ; and for engines using steam at low or medium pressure governed by throttling, it will hardly be superseded.

STEAM-PORTS.

When designing slide-valves, one of the first data required is the size of steam-ports. No general rule can be given for this ; in fact, progressive engineers consider it an unsettled matter. Much depends on the piston speed, but there are other influential conditions. If it is desired to admit steam to the cylinder under full pressure, larger ports are required than if the pressure is to be throttled down ; and if a considerable decrease of speed is permissible under an occasional extra heavy load, as, for instance, by hoisting-engines, comparatively small ports give entire satisfaction, and are preferable to larger ones. The length and shape of steam passage and port may have an important bearing on their sectional area, but narrow ports are requisite in order to keep the valve motion within practical limits. The discovery may still be made that the passage, and not the port, is the governing element. The "mechanical instinct" of the engine-builder will make the port conform to the passage, but this instinct is not reliable when the molecular action of steam is involved. In the writer's personal experience, for instance, an abrupt contraction of the port to less than half its original size, as shown in Fig. 8, did not

* See page 42.

lower the admission-line or raise the exhaust-line on the indicator-card. This phenomenon is susceptible of a rational explanation, but mere speculation is not in place here.

Incidental experiences seem to prove that the condition of the steam, whether wet, dry, or superheated, is of no little consequence. Dry steam, being more subtile than wet steam, moves easier or does not adhere to the sides, and therefore does not require so large ports or passages.

Fig. 8.

It may safely be assumed, that if the ports and passages are large enough to allow the exhaust to pass through without appreciable back pressure, they are amply large for the entering steam, and if the indicator card shows a "good" exhaust-line, the ports are large enough, no matter how bad the "steam-line" appears.

For some years past it has been a rule to make the ports of high-speed engines large enough to make the mean velocity of the steam current 6,000 feet per minute, which for 600 feet piston speed makes the port area equal to one-tenth of the piston area; but occasional experiences indicate that, if the passages are short, a much smaller area may give perfect satisfaction, and the above rule is probably now becoming obsolete.

The rule here mentioned entirely ignores friction against the walls of the passage; this could be remedied by introducing some power of the diameter of the cylinder as a separate factor in the formula. The general formula might, for instance, be: Velocity of steam, in feet per minute, equals 3,000 times the cube root of the diameter in inches. For an 8-inch cylinder this would make the mean velocity through the ports 6,000 feet per minute, and for a 27-inch cylinder it would make it 9,000 feet.

PORT—OPENING.

"**Port-opening**" means the uncovered part of the port, and it should not be confounded with the "**port,**" which is the orifice

of the steam passage. The valve may not open for steam as much as the width of the port; for experience has taught that restriction of a steam passage by two sharp edges, within a certain limit, has very little, if any, effect on the steam current. It is known, for instance, that a throttling-valve of the piston type does not have to open but very little to allow full pressure steam to pass through, and on many automatic cut-off engines, the port opening is much restricted without appreciable loss of pressure.

VALVE DIMENSIONS.

After the travel, point of cut-off, and lead have been determined upon, a circular diagram like Fig. 7 may be drawn, and this will give the lap and port opening. If a larger port-opening is desired the lap and travel must be increased proportionally. A line may be drawn obliquely across the parallel lines as shown, so as to give the desired port-opening, and the other dimensions will appear at once, correctly proportioned, on the same line. By this method the lead will be increased with the port-opening, but the **period** of preadmission remains unaltered. In any case, it is advisable to draw a fresh diagram for any contemplated alteration, for the labor it involves is very insignificant compared with resulting clearness and simplicity; and the final diagram should be duly labeled and kept as a record for future reference.

After port, travel, steam- and exhaust-lap have been obtained, a section of valve and valve-seat may be drawn; and it should be observed that the exhaust after leaving the port is not interfered with too much by the opposite exhaust edges. On unbalanced valves it is advisable to reduce the exhaust cavity as much as possible; and in that case, the central exhaust opening at the extreme of the travel may be reduced to five-eighths of the area of the steam-port by overlapping of the valve.

VALVE DIAGRAM FOR SHIFTING—ECCENTRIC ENGINES.

By making the parallel lines in Fig. 7 slant the other way, as if the figure were turned end for end, we get a diagram which has a certain constructive advantage and also a less desirable feature. It becomes a left-hand diagram for a right-hand crank motion, as it were; and the various valve movements, incidental to the turning of the crank, must be traced out from right to left, while the crank actually turns from left to right. But such diagram will present the slanting lines at right angles to the eccentric arm, which facilitates its construction for valves with variable travel, as used on certain automatic cut-off engines and on locomotives and marine engines.

Fig. 9 shows how to lay out a valve diagram in this manner for an engine having two valves driven by separate eccentrics, one of which controls the steam admission and the other controls the exhaust. The induction-valve is driven by an eccentric of variable throw, the center of which is supposed movable, relative to the crank, on the curved path ABC, whereby its **throw and angular advance are simultaneously changed.** Thus, when the period of admission is shortened by reducing the throw, **the proper lead is maintained by a corresponding angular advance.** In the natural diagram, Fig. 6, lead is represented by the horizontal distance of the center of eccentric from chord 1–4, and it can readily be reproduced in Fig. 9 in

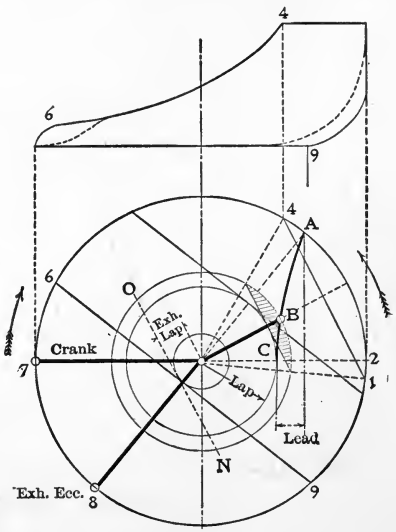


Fig. 9. Sweet's Valve Diagram.

the following manner: With a radius equal to the lap draw a circle, as shown, and drop a vertical from center of eccentric; the distance from this to the lap circle is lead. In the present case, the lead becomes less as the eccentric advances, and is nil at C, opposite the crank. When the eccentric is at full throw, the diagram becomes an exact counterpart of Fig. 7 in a reversed position; but suppose the valve were to cut off steam at one-quarter of the piston-stroke, **without changing its travel,** then the chord 1–4 would become short and steep, as shown, and the eccentric would be advanced to a position at right angles to this chord, and the lap would have to be increased, as represented by the increased distance of the chord from the center. But as the lap of the valve cannot vary, its travel must be reduced in order to obtain the requisite proportion between lap and travel; and this is effected by moving the eccentric from A to point B, where a tangent to the lap circle, at right angles to the eccentric-arm, will cut off an arc of the reduced circle of rotation through B, which arc, or its center angle, represents period of admission within the prescribed limits for one-quarter of the piston-stroke. The small shaded segment shows the port-opening, which is remarkably small considering that such engines often carry full load when the valve is cutting off at that point in the stroke.

It should be noted that the velocity of the piston varies — it is greatest at mid-stroke and becomes nil at the ends — and that the efficiency of the port-opening depends on the local speed of the piston. For this reason only a very small opening is required for admission of steam at the beginning of the stroke; but at one-quarter of the stroke the piston-speed has greatly increased, and in order to get a good "steam-line" and a "square cut-off" there should be ample steam-opening near the point of cut-off. Fig. 9 shows clearly how unfavorable the conditions are for a square cut-off early in the stroke of single-valve "automatic engines" and locomotives, and how incom-

parably better the conditions become by later cut-off. An early cut-off by a single valve must, in any event, reduce the port-opening considerably on account of the long lap required; but by automatic or adjustable cut-off, it becomes still smaller, on account of the reduced valve travel. It is also worthy of notice that the rapidity of the valve action is less by early cut-off, because this occurs when the eccentric is near the extreme of its travel, where its lateral motion is comparatively slow. The port opening, when steam follows one-quarter of the stroke is, in the case here considered, about one-sixth of the maximum opening, or about three-sixteenths of an inch for a 12-inch cylinder.

The exhaust eccentric is supposed to be fixed on the shaft, giving a constant valve travel and fixed points of release and exhaust closure, as in Fig. 7. With direct valve connection and exhaust escaping over the inside port edges, the exhaust eccentric would be in position A; but in the case under consideration, the exhaust is supposed to take place over the outside edges of the ports, and the eccentric will, therefore, be at point 8, diametrically opposite point A.

If steam and exhaust valves were driven by one eccentric, the slanting lines in the diagram would be parallel, and when steam is cut off at one-quarter stroke, the exhaust opening and closure would be at O and N respectively; which means premature release and much compression. By using two eccentrics the early release is avoided, and the compression, being constant, is better adapted to promote smooth running under all conditions. Lap may be used to advantage on a separate exhaust-valve.

Note that in this diagram the slanting lines are, as a rule, **not parallel.**

It should also be noted that the opening for exhaust is always ample, and takes place when the eccentric is at or near its half-throw, and that, therefore, an early release is not required.

As here represented, the fixed travel of the exhaust-valve is

equal to the maximum travel of the steam-valve; but this is evidently not necessary, as a much shorter travel of the exhaust-valve would give ample opening for release. When a single valve controls both steam and exhaust, the exhaust lap is usually omitted; and though the exhaust opening is much reduced by the reduced throw, it is still ample when steam is cut off at one-quarter of the stroke; and as the exhaust lead is nearly constant, the very early release must be detrimental to steam economy.

The ideal indicator diagram, annexed to Fig. 9, illustrates the action of an automatic admission and cut-off valve in combination with a separate exhaust-valve, and dotted lines show the corresponding action of a single combination-valve, the point of cut-off in both cases being at one-quarter of the piston-stroke.

If preferred, the diagram Fig. 7 may be used for shiftable eccentrics. It will only be necessary to draw the shifting-path of the eccentric on the other side of the center, where it is to be considered as a construction line only.

VARIATION IN PORT-OPENING.

Diagram 10 represents variation in port-opening. When steam is cut off at one-quarter of the piston-stroke, the variation is represented by consecutive ordinates in the shaded areas, and the light areas show the variation when steam is cut off a little before three-quarters of the stroke. Admission and exhaust are here supposed to be controlled by a single valve, and the exhaust-opening is represented by ordinates to the lower curve. Ordinates to the crank-circle represent — on the proper scale — velocity of the piston, and it will be noticed that at one-quarter stroke it has nearly attained its maximum velocity.

For a given point of cut-off, a certain relation exists between lap and travel, and for earlier cut-offs the lap becomes greater, or else the travel becomes less. If the lap for a given cut-off is increased, the travel must increase at the same rate or nearly

so; and the port-opening, bearing a fixed relation to the lap and travel, is increased proportionally. Hence, by increasing the lap of a valve of variable travel, the port-opening is also increased; but the increased lap curtails the range of cut-off, for the latest cut-off is determined by **the relation of lap to maximum travel.** In this connection it is sometimes stated that the port-opening is proportional to the lap, which is absolutely true only if the lead is zero, or if it is directly proportional to the travel.

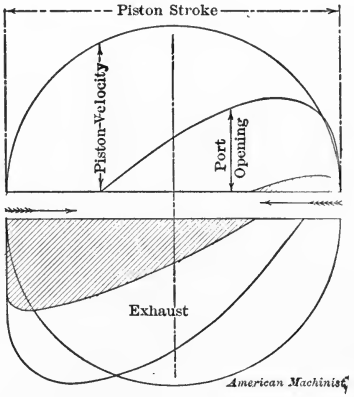


Fig. 10.

A wide range of power is often desired in automatic cut-off engines, 25 to 35 per cent above the rated or economical load is specified, and the valve and governor must be designed to meet this requirement. An extreme cut-off at three-quarters of the stroke is sometimes imperative; but at the same time, it should be kept in mind that there is a certain desirable or practical limit to the travel of the valve, and that by extending the range of cut-off the port-opening for earlier cut-offs becomes smaller. Let the travel be four inches, and let the latest cut-off be at three-quarters of the stroke, then when cutting off at one-quarter stroke, the port-opening is about three-sixteenths of an inch; but if latest cut-off is at five-eighths of the stroke, the port-opening for one-quarter stroke becomes about one-quarter of an inch; and making one-half the stroke, the limit for cut-off increases the port-opening for one-quarter stroke to five-sixteenths of an inch.

NOTES ABOUT LEAD.

It used to be general practice to give the valve a constant lead, irrespective of the point of cut-off; but the theory in sup-

port of this practice is now fast becoming obsolete, and later experience has established the fact that the lead may vary considerably without detriment to smooth running.

Lead means width of port-opening at commencement of the stroke; and no reference is made to the **period of lead** or **lead angle**, as represented by arc 1–2, or by the intercepted angle, though it seems quite proper to take this — the time element — in account. Assuming a constant lead, the lead period will vary according to the location of the point of cut-off. By earlier cut-off the steam is admitted earlier to the cylinder, and it has consequently more **time** to fill the clearance spaces; it will be more effective on that account, and it can hardly be disputed that this is of as much consequence as the extent of port-opening at the end of the return stroke — or perhaps more so.

If in Fig. 9, the path ABC is made straight and perpendicular, the lead will be constant; and any desired variation in lead may evidently be obtained by making the eccentric shiftable in other directions, more or less deviating from the perpendicular. In the particular case represented in Fig. 9, the lead decreases to nothing at the point of minimum throw, and when the eccentric is near this point, the port-opening becomes exceedingly small; but if the valve has some lead at this point, it will become the minimum port-opening, and it would make a **comparatively** large addition to the port-opening by very early cut-off, and less "wire-drawing" of the steam would probably be the result; but according to later theories, wire-drawing under these conditions is beneficial; for, by lowering the initial steam pressure, and by rendering somewhat drier steam, it reduces the initial condensation in the cylinder; and it is claimed that in case of a single-valve engine early admission is unnecessary in any event, because an early exhaust closure, incidental to early cut-off, fills the clearance space with compressed exhaust steam.

It is also easily proven that by curtailing the lead, the angular advance is diminished, and release and exhaust closure occur correspondingly later in the stroke.

Negative lead in conjunction with short cut-off has the undoubted practical advantage that it will always insure perfect control of the speed of the engine when running light, whether the valve is set exactly central or not; and it becomes a particularly useful expedient when the engine is connected with a condenser.

If there be any positive lead, the clearance space will fill with full-pressure steam before the commencement of the stroke, and the expansion of this steam may run the engine if the back pressure is light. Much depends on the clearance space. Let it be 8 per cent of the piston displacement, and let the initial absolute pressure be 115 pounds per square inch; then the **absolute** mean pressure during expansion of the clearance steam will be 23 pounds; but, if the clearance were 4 per cent of the displacement, the absolute mean pressure would be 15 pounds only. The frictional load is a variable or unknown quantity, and it may be very small when the engine is directly connected with a dynamo. With atmospheric back pressure and heavy compression there is little danger of the engine "running away," but when condensing the conditions are quite different.

COMPRESSION.

It should be noted that the height of the compression curve varies inversely as the clearance, and that, therefore, early exhaust closure is compatible with a large clearance space, while it may be objectionable in connection with small clearance.

In single-valve engines and locomotives the work done in the cylinder is partly regulated by the variable compression, and the variation of the point of cut-off is therefore less than where a fixed eccentric governs the exhaust, and the excessive wire-drawing of steam incidental to very early cut-off is thereby partly avoided. Otherwise there is nothing gained by variable compression; for, according to D. K. Clark, the absolute loss by initial condensation is nearly constant for all points of cut-off up to

35 per cent of the stroke; and the ratio of useful work to waste, or the cylinder efficiency at early cut-off, depends therefore entirely on the work done per stroke of engine, irrespective of the point of cut-off.

The height of the compression curve in the indicator diagram depends on the point where compression commences, on the clearance space, and on the absolute back pressure or density of the exhaust — the greater the back pressure, the higher the curve. And as it may reasonably be assumed that the most economical compression is that which fills the clearance space with steam of nearly same density as the entering steam, the pressure of this should also be taken into account.

From this point of view it appears that under certain conditions the single combination valve may become the ideal valve, and that such conditions may exist in the high-pressure cylinder of certain compound engines, where the receiver or back pressure varies according to the point of cut-off, being greatest with late cut-off and late exhaust closure, and diminishing as the point of cut-off and exhaust closure advances; for this may, under the conditions imposed by a single combination valve, lead to a nearly fixed condition of density of the compressed steam.

Judging from the above remarks, it will readily be agreed that a fixed rule for the determination of travel, lap, lead and port-opening may become worse than useless on account of our deficient knowledge of what is required for best economy, and that a wide scope may be given to practical considerations.

THE MOTION OF LOCOMOTIVE VALVES.

To construct a locomotive valve diagram, it is only necessary to substitute for the link-gear an equivalent single shiftable eccentric; that is, to find the path of the center of a shiftable eccentric which will produce the same valve-motion as does the link.

Fig. 11 represents diagramatically the chief mechanism of a stationary link at mid-gear, and the positions of crank and eccentric centers are indicated. The virtual angle of advance, which should be used in the valve diagram, is a little greater than the actual angle; and the diagram shows how to obtain it.

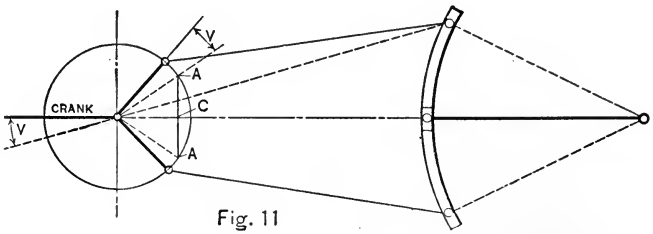


Fig. 11

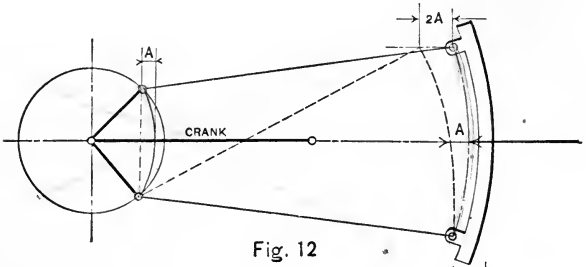


Fig. 12

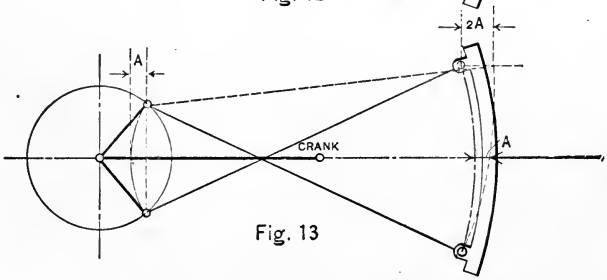


Fig. 13

The lead is constant, and the straight line ACA represents the path of the equivalent shifting-eccentric; but the distance AC is sufficient for the valve diagram.

Diagrams 12 and 13 represent "shifting-links" with "open" and "crossed" rods respectively. The valve is supposed to

be moved by a counter-arm rocker, which is not shown; and this requires the eccentrics to be placed on the crank-side of the shaft, or diametrically opposite the position they would take if there were no intermedial reversal of the link motion. In full gear the motion is supposed to be governed entirely by one of the eccentrics, that is, when the throw of the link-block equals the throw of the eccentric; but in mid-gear the motion is governed by the two eccentrics jointly, and if the angularity of the rods were the same for both dead-center positions of the crank, the mid-gear throw would be exactly as that of a single eccentric located at the center of a straight line joining the two eccentric centers, and the lead of the valve would be constant for the entire range of cut-off. But, owing to the fact that the angularity of the rods is not the same on both centers, the mid-gear valve motion will be increased with open rods and diminished with crossed rods, and the mid-gear lead will be increased or diminished proportionally. The shifting-path of the single equivalent eccentric will, therefore, be curved as shown in the figures.

If it is desired to equalize the lead for both ends of the cylinder, the link must conform to a certain curve; and any variation from this curve will make the lead unequal for back and forward centers for all points of cut-off but one, and this point may be fixed at will by lengthening or shortening the valve-stem connection.

How to lay down the shifting-path of the equivalent single eccentric is shown in the diagram, and it is also shown how to locate the central point in the link arc, so as to give uniformly equal lead at both cylinders ends. The letter A denotes a certain distance or unit measurement, and its designation in the figures makes any textual explanation unnecessary. The eccentric-rod is supposed to be directly in line with the rocker-pin in two full-gear positions.

The rocking motion of the link may cause a marked dis-

placement of the point of cut-off, if the link and saddle pins are not properly located; but as this involves the valve-gear only, it will not be discussed here.

The shifting-link with open rods is most commonly used on American locomotives; and it is noteworthy that this gear provides maximum lead in combination with early cut-off, which is the reverse of stationary engine practice. The heavy compression in mid-gear may reverse the strains gradually, while, with light compression in full gear, and at slower speed, excessive lead may cause a violent reversal of the strains.

Variable lead is unavoidable with the shifting-link, but the lead may be reduced to any extent and made partly negative by providing sufficient lap or by moving the eccentrics back.

MULTIPORTING.

After Corliss had established the superior economy of early cut-off, and by his peculiar valve-gear had produced a nearly sharp cut-off at moderate speed, it became almost an axiom that a sharp cut-off is essential for best economy; and it has apparrently been the object of engine-builders, ever since, to produce an indicator-diagram with "square cut-off" — which means a nearly horizontal "steam-line" terminating abruptly where it joins the expansion line.

The slanting steam-line and round corner is the result of "wire-drawing," which is another name for **free expansion**; but it is also due to the **working expansion** of the steam already in the cylinder. While the free expansion, in one sense, represents a direct loss, it may, at the same time, have a beneficial effect, if the steam is not absolutely dry; and it is, therefore, just possible that the slanting steam-line and the round corner does not represent any actual loss.

No actual knowledge, however, can result from mere reasoning in this case; but the question might be settled by careful testing, which to the writer's knowledge has never been attempted.

In order to obtain sufficient port-opening with reduced travel, many valves have one or two supplementary steam-passages, which communicate with each steam-port and into which steam is admitted over special steam edges, which open and close for steam admission simultaneous with the main steam edges.

How much this actually improves the steam-line on the indicator card has probably never been fully investigated, or experimental results have not been published. Such experimenting would be extremely simple ; for the supplementary ports in the valve could easily be temporarily blocked, and it may have been tried in some university laboratory.

Common sense suggests that two port-openings will give a better steam-line and sharper cut-off than one opening, but nothing definite is known about it. Many incidents in steam-engine practice may be accounted for by the hypothesis that a sharp local obstruction in the steam-passage has an almost imperceptible effect, until the free opening has been reduced to a certain extent, after which a rapid decrease in the flow follows ; and if that be so, the next pertinent question is: How much may the opening be reduced before this critical point is reached ?

In this connection, the fact should be recognized that the indicator diagram is not a reliable criterion on steam economy ; also that a valve which will stay comparatively tight is a requirement of prime importance, for whatever the loss or gain by wire-drawing, compression, release, etc., the peculiar steam distribution due to leaking of the valve can under no circumstances be profitable ; and as a multiplication of the port-openings may increase the inevitable leakage loss, it is quite evident that if no objection could be raised to the round corner and slanting steam-line on the indicator card, many valves would be reconstructed.

SETTING THE ENGINE ON DEAD CENTERS.

When the piston is at the end of its stroke, and the connecting-rod is right on the center line of the engine, in line with

the crank-shaft, then no amount of steam pressure will put the crank in motion, and the engine is therefore said to be on its dead center. When near the dead point the piston-motion becomes very slow, and it is actually reduced to nothing at the moment when the motion is reversed. Near the end of the stroke the crank-motion is nearly at right angle to the piston-motion, and it becomes actually so at the moment the crank passes the center line. At, or near, this point there is very little motion of the cross-head — it becomes nearly stationary — and it is, therefore, very difficult to determine the dead center position of the crank by observing the motion of the cross-head. On the other hand, the eccentric is not far from mid-throw; and the slide-valve is, therefore, moving rapidly at that point; and in order to set it to its proper lead at the commencement of the stroke, the exact dead-center position of the crank must be ascertained. It may be done in the following manner:

Place crank near dead-center, and make a mark at the edge of the wheel by placing a tram, or stick, against the floor, or some other fixed object near the rim of the wheel; also mark position of cross-head; then turn engine over dead center until cross-head returns to position first marked; mark rim again with same tram, in same manner. Now put a mark on rim centrally between the two tram-marks, and turn wheel till this mark coincides with tram-point. This is the exact dead-center position. Locate the opposite dead center in exactly the same manner, and set the valve to give equal leads. If there is lost motion in the brasses it is not possible to determine the stroke of the piston exactly, but any material error in the valve-setting may be obviated by turning the crank in one direction only.

CHAPTER II.

IMPROVED SLIDE-VALVES.

THE DOUBLE-PORTED MARINE SLIDE-VALVE.

PLAIN double-ported slide-valves, constructed as shown in Fig. 14, are often used on marine engines. The valve is shown in its middle position, and it will be observed that the two conjugate ports are exactly alike, and that they are opened and closed simultaneously. Transverse tapered passages in the valve admit steam from the steam-chest to the inner ports, and

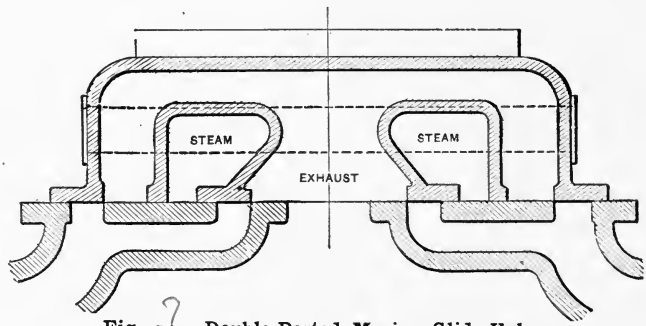


Fig. 14. Double-Ported Marine Slide-Valve.

a central duct, cored in the valve, conveys the exhaust from the outer ports, over the transverse steam-passages to the common central exhaust opening.

In marine engines the intermediate and low-pressure valves are not used for short cut-off; but as the throw is usually the same for all the cylinders, and therefore proportionately short for the large cylinders, double ports become desirable. In such cylinders the unbalanced pressure is moderate, and unbalanced valves with short throw work satisfactory.

For the smaller cylinders — where smaller valves are required, and the throw is comparatively greater, and where the differential pressure occasionally becomes considerable, and where the high temperature makes lubrication less effective — single-ported piston-valves are generally used.

The double ports necessarily increase the clearance and the cooling surface; but in large cylinders, where steam is cut off near half stroke and extreme variation in temperature is avoided, this becomes of less consequence.

Quite a number of special valve-gears are in use, by which a quick and ample port opening is obtained with early cut-off, and thus one of the objections to single-valve variable cut-off is removed; but, as previously explained, the evil of premature release and variable compression are attributes of the single valve, and cannot be amended or ameliorated by any simple valve gear.

THE ALLEN LOCOMOTIVE VALVE.*

As representatives of the double admission type, the Allen locomotive-valve and the Straight-Line balanced valve are shown in Figs. 15 and 16.

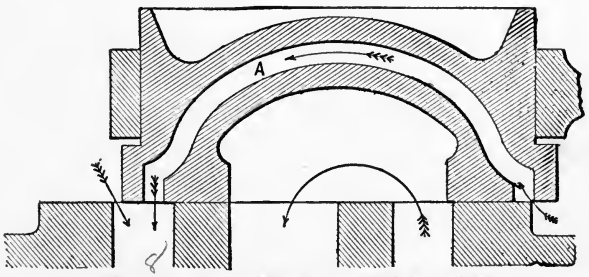


Fig. 15. The Allen Locomotive Valve.

In the Allen valve, Fig. 15, a single, long, double-ported steam-passage, A, takes steam over supplementary steam edges, so located as to open and close the entrance opening of the pas-

* This valve is called the Trick valve in Germany, after the German inventor.

sage simultaneously with the opening and closing of the main port. This passage never communicates with the exhaust; for its outlet to the main port is closed just before the port opens for release, and it is opened just after the port is closed for the exhaust, and it is then filled with compressed exhaust steam. By short travel the heavy compression fills it with dense steam, and it ought to serve its purpose well; but when the travel of the valve is increased under heavy load, it becomes of no use, and there will then be a small loss by filling it with live steam at the commencement of each stroke.

THE STRAIGHT-LINE BALANCED VALVE.

The Straight-Line valve, Fig. 16, is the pioneer, and embodies all the principal features of the class of valves it represents. It is a double-faced valve, the two opposite faces being exactly alike. It is confined between the cylinder valve-face and an exactly similar face of the stationary "pressure-plate" C, which plate carries the unbalanced steam pressure that would

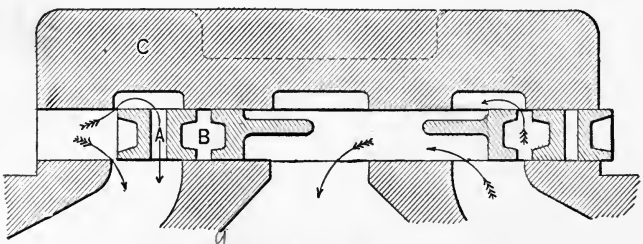


Fig. 16. The Straight-Line Valve.

otherwise be carried by the valve. This pressure plate is held in its proper position by distance pieces, which allow the valve to move freely between the opposite faces. This construction provides two pairs of steam edges at each end of the valve, and two port openings are obtained, as shown by the arrows. The passage A conveys the steam from the shallow recess in the pressure plate to the main port, and it operates exactly as

the corresponding passage in the Allen valve. It is therefore desirable to restrict this space as much as is possible without destroying its usefulness. In case of the flat balanced valve, this presents no practical difficulty ; but such narrow passage could not easily be cast in an Allen valve.

The recesses in the pressure plate are not merely steam-passages, they are also there to equalize the pressure on both sides of the valve. They may be considered as extensions of the main steam-passages and as additions to the clearance space ; and being half the time exposed to exhaust steam, the additional surface they expose to the entering steam is one of the less desirable features of this and all other balanced valves.

The main object with the exhaust passage B is to get a "quick" opening and closing of the exhaust, so as to avoid "wire-drawing." After the exhaust is cut off part of it is compressed in this space, and is bottled up there before live steam enters the port ; and directly after this is cut off the bottled-up steam is allowed to mingle with the expanding steam in the cylinder.

If a considerable quantity of water is carried over in the cylinder it may fill the clearance space, and force the valve and pressure-plate from their seats, while the water is discharged in the steam-chest ; and thus the cylinder is relieved of a dangerous strain, while the clatter of the valve attracts the attention of the engineer.

This valve is used in connection with a shaft governor, which automatically changes the throw and angular advance of the eccentric by shifting it across the shaft.

An automatic induction valve of the Straight-Line type is sometimes used conjointly with an exhaust valve of the same type actuated by a permanent eccentric. These valves are placed on opposite sides of the cylinder, and are similar in construction to the combination valve shown in Fig. 16. The

main object with this arrangement is to have constant release and compression, as explained in the first chapter.

The exhaust pressure-plate must be held to its seat by mechanical means ; and it may be done by springs, in which case it affords a direct escape for entrapped water to the exhaust pipe.

It is questionable whether a double exhaust opening is useful, or even desirable, if the valve has no exhaust lap ; for in that case, it opens and closes for the exhaust when the eccentric is at "half throw," which is the most favorable condition for rapid action ; and as the exhaust opening for such valve is equal to lap plus lead, irrespective of the throw, when the opposite steam-port opens, or practically at the end of the stroke, the early release cannot be necessary ; and it is a natural conclusion that a slow opening will be advantageous under such conditions. The exhaust closure cannot be too "quick," but it is a question if the loss by quick opening is not greater than the gain by quick closing. With late admission and some exhaust lap, two exhaust openings may become desirable.

VARIATIONS OF THE STRAIGHT-LINE VALVE.

The valves and pressure-plates of the Straight-Line type are sometimes so constructed that they cannot be forced from their seats, and these valves admit steam through a central opening, and exhaust it over the outside edges ; which leaves exhaust pressure in the steam-chest and makes packing of chest cover and stem easy, and it also allows examination of the valve with steam-pressure on. **The transposition of steam- and exhaust-edges brings the eccentric in a diametrically opposite position on the shaft.**

Some valves of the Straight-Line type open and close for steam simultaneously in four places, in which case two extra pairs of steam edges on opposite valve-faces are disposed much like the two supplementary steam edges of the Allen valve, and

this necessitates one or two steam-passages lengthwise through the valve body. By reason of the double face twice as many openings are obtained as with the single-faced Allen valve.

THE McEWEN VALVE.

A slide-valve is supposed to be in perfect equilibrium when the steam-pressure is equally distributed on opposite faces, neglecting the weight of the valve, which is generally of little consequence. This condition is presumed to be satisfactorily

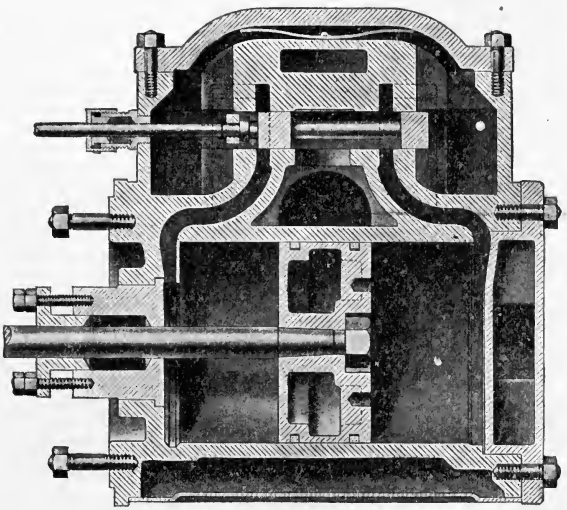


Fig. 17. **The McEwen Valve.**

fulfilled when there are channels or recesses, back of the valve, in form and area equal to, and exactly opposite the ports and the exhaust area in front of the valve, and means provided for continuous communication between opposite sides. It must, however, be admitted that, under the conditions which generally

prevail, the intensity of the pressure on both sides of a balanced valve cannot be absolutely uniform; for when the ports are only partly covered by the valve, and steam is rushing through the ports with a velocity of one hundred feet, or more, per second, there is practically no pressure on that surface which overlaps the port edge, while the pressure on the corresponding opposite surface, on the back of the valve, must of necessity be considerably greater, partly on account of the dynamic effect of the steam, and partly from the resistance the flow of steam encounters in the passages back of the valve.

This undesirable feature is avoided in the design of the McEwen valve, shown in section in Fig. 17. The valve is of flat, rectangular form, and is of the "Straight-Line" type; but, unlike Professor Sweet's valve, it has no auxiliary passage through the valve. The valve is covered by a pressure-plate, in which are auxiliary steam-passages, the ports of which are opposite to the ports in front of the valve, and being simultaneously uncovered by the valve, these ports permit the steam to flow in opposite directions from opposite valve-faces, thus leaving the valve in perfect equilibrium.

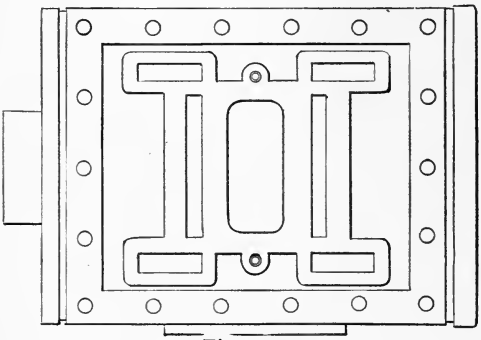


Fig. 18.

The valve-face on the cylinder is shown in Fig. 18, and it will be understood that the pressure plate has four hollow legs, by which steam is conducted through the four smaller ports into the main steam-passages to the cylinder; and the exhaust will escape through the same ports.

THE BALL TELESCOPIC VALVE.

The excessive unbalanced pressure on the common D-valve, which causes friction and rapid wear, is mainly due to the large exhaust cavity which permanently removes steam-pressure from a large area on the face-side of the valve, while the entire rear side is exposed to full steam-pressure. By admitting steam through a central opening in the valve, and exhausting it over the outside edges, the exhaust cavity is dispensed with, and the unbalanced pressure is thereby greatly reduced.

A valve designed on this principle by Mr. Frank H. Ball is represented in Fig. 19, which shows the valve in its central

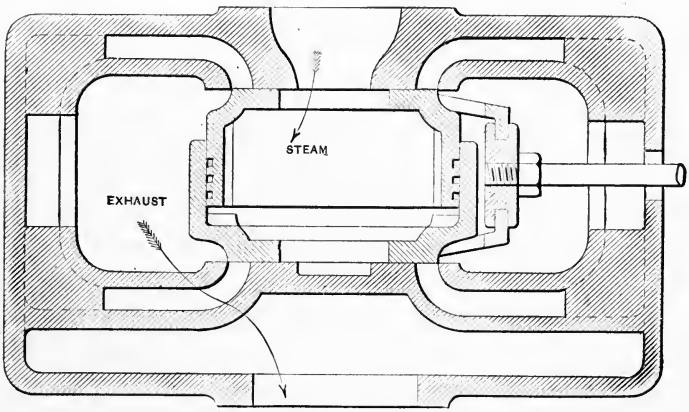


Fig. 19. The Ball Telescopic Valve.

position in the steam-chest. It is a double-faced valve, and it consists of two telescopically connected parts. Each part consists of a flat rectangular frame, which covers the ports in its central position, and on which is a short hollow cylinder. One of these cylinders fits inside the other, and it has three grooves containing packing-rings. As these rings do not wear at all they remain steam tight when properly fitted. Steam is admitted to the inside of the valve, as shown, and the exhaust

escapes over the outside edges into the steam-chest. The only unbalanced area is that portion of the steam-ports which is opposite the cylindrical part of the valve during the exhaust period; and taking the counter pressure into account, the valve is so proportioned as to leave sufficient unbalanced pressure to insure a close contact between the working faces. This valve has the undoubted advantage that it will follow up its own wear; and the claim made for it, that it remains steam tight throughout its entire life, does not seem unreasonable; and, moreover, this claim is backed by seventeen years' experience.

The double-port opening is particularly useful when the steam is cut off early by reduced valve travel; but it will be noticed that the valve-faces are horizontal, or parallel with the top and bottom of the steam-chest, which necessitates a divided and somewhat distorted steam-passage to the cylinder, and is a feature which some designers studiously avoid.

BALANCING A COMMON D-VALVE.

The extreme changes of temperature in cylinder castings may warp the valve-faces; and for this and other reasons a plain slide-valve, provided with a self-adjusting pressure-relieving device on its back, is often used. As the surfaces are self-adjusting extraordinary nice fitting is not essential, and if there is a leakage it is apt to diminish by wear; and if much water gets into the cylinder it may escape to the steam-chest by pushing the valve back from its seat. Such a valve cannot be perfectly balanced; for the counter-pressure on the face depends to some extent on the position of the valve, and is, therefore, not quite uniform.

Fig. 20 shows a plain slide-valve with a pressure-relieving device suitable for a horizontal low-pressure cylinder. It consists of a ring, cast on the back of the valve, and a loose flanged ring, fitted inside it. The ring bears against the steam-chest cover and keeps steam away from the back of the valve.

The space inside the ring must communicate with the exhaust, and in the case here illustrated the circular opening extends through the back of the valve to the exhaust cavity. Small helical springs are placed so as to keep the ring out when the engine is started.

The valve here shown was designed for the low-pressure cylinder of a non-condensing engine and for a fixed cut-off at three-eighths of the stroke. To prevent the compression from forcing the valve out against **low receiver-pressure** the ports were reduced to about half their usual size, as shown; and, as mentioned in chapter 1, this had no appreciable effect on the exhaust. The relieved area is that inclosed by the outside circumference of the inside ring or the inside circumference of the outside ring, and for a non-condensing engine it should not exceed three-quarters of the area between exhaust edges. For condensing engines it could and ought to be nearly equal to this whole area; but as all engines may incidentally run non-condensing, it is probably advisable to provide for such occurrence.

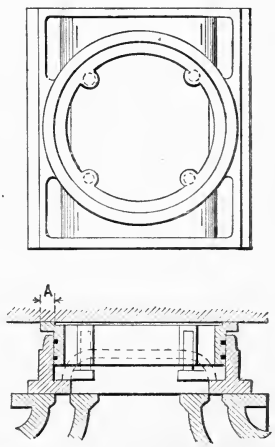


Fig. 20. Plain Valve, Partly Balanced.

On high-pressure cylinders the ring area is often made equal to the area between exhaust-edges plus the area of one port.

The pressure that keeps the ring against the steam chest cover is determined by the difference in pressure on both sides of the flange A. Professor S. W. Robinson has shown experimentally that a sliding surface which separates exhaust from "live steam" is exposed to pressure of "creeping" steam, which decreases nearly uniformly from the steam side to the exhaust side; and it may reasonably be assumed that, when one surface

is sliding over another surface which is alternately exposed to steam- and exhaust-pressure or high- and low-pressure steam, the mean pressure between the surfaces will be a mean between the extremes on either side; but it may also reasonably be expected that, by the reciprocating motion, the intervening pressure becomes somewhat uneven at the two extremities of the travel; for at one end the covered surface is never fully exposed to the lower pressure, and at the other end it is never exposed to the higher pressure, and in the extreme positions the intervening pressure may, therefore, alternately approach the higher and lower pressure at diametrically opposite points of the ring. On one face of the flange is full receiver-pressure, and on the other face is the variable intervening pressure; and it will readily be understood why it has been found necessary, in order to insure permanent contact, to make both flange areas equal by turning down a shallow recess, as shown. The flange A can be from three-quarters to one inch wide, and the springs should be very light. Similar devices are often used on locomotives.

THE RICHARDSON BALANCED VALVE.

The Richardson balancing device, much used on locomotives, consists of straight metal strips, which fit steam-tight in grooves on the back of the valve, and are held against the steam-chest cover by steam-pressure and springs. These strips inclose a rectangular area which communicates with the exhaust cavity through a small opening in the back of the valve.

THE THOMAS BALANCE.

Fig. 21 represents a balancing device invented by Mr. W. J. Thomas and much used on locomotives. The ring A is open in one place and is expanded some by being forced on its conical seat; it is therefore self-adjusting, both against the conical surface and against the planed surface of the steam-chest cover, and the steam-pressure on the circumference of the ring insures

a tight joint between the ring and the cone. By using the proper taper, the resultant pressure against the steam-chest cover is made sufficient to overbalance the counter-pressure of creeping steam between the ring and the cover. The differential pressure on the valve-face is determined by the relative proportion of the relieved area. In locomotives due allowance must be made for removal of the valve-yoke, and, therefore, a removable disk is used, as shown in the cut. If the steam-chest is narrow, two rings may be placed side by side on cones cast on the valve.

The manufacturers have established the following rule: Balance-ring areas are made equal to "area of one steam-port, two

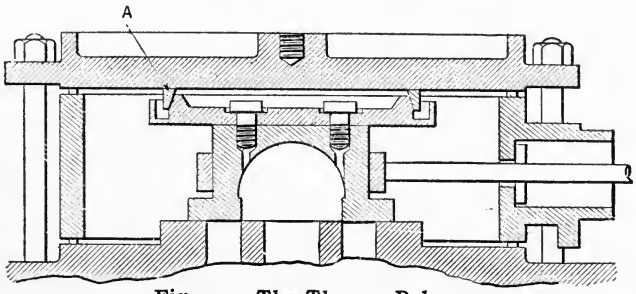


Fig. 21. The Thomas Balance.

bridges and the exhaust-port plus 8 per cent if for single balance, and plus 15 per cent if for double balance." It may appear as if more pressure is removed from the back of the valve than would ordinarily be considered safe, but the apparent anomaly disappears when it is considered that the reaction from the bevel ring-surface must be balanced; and the friction of the conical joint, no doubt, serves a good purpose by opposing the counter-pressure of the exhaust-steam when the link is at mid-gear. The opening, cut in the ring in order to expand it, is covered by an L-shaped joint-plate, which fits against the cone and is flush with the top of the ring, and forms a steam-tight joint in both places. The rings are made of hard, close-grained cast iron;

and the outer rim, shown in the sketch, is there to prevent accident if the ring should break. As the ring is held firmly against the cone, there is no wear on this; and the ring, being self-adjusting, is admirably adapted for the interchangeable system prevalent in locomotive works. This device may readily be fitted to any common slide-valve or double-admission valve.

That more balance area is required with double rings is evidently due to the fact that the circumference of two rings combined is more than that of one ring inclosing the same area, and the push against the valve-seat from the conical ring surfaces becomes correspondingly greater.

PISTON-VALVES.

Piston-valves are cylindrical valves moving in the direction of their axis. Usually steam is admitted centrally, and the steam-ports open into annular spaces surrounding the valve, by which a perfect balance is obtained. They are simply balanced slide-

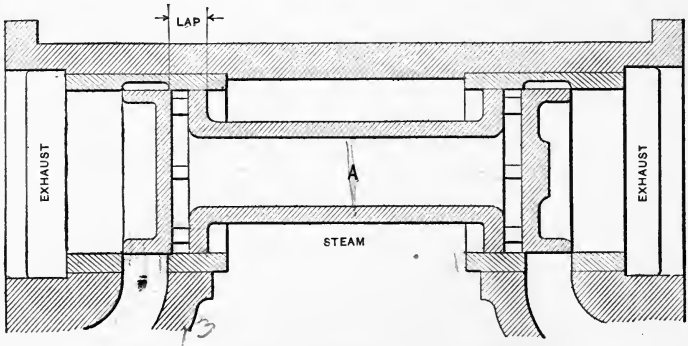


Fig. 22. Double-Admission Piston-Valve.

valves of convenient form, and are designed for single or double admission on the same principle as the flat valve. Fig. 22 shows a double-admission piston-valve in its central position, with steam-laps on the inside between the ports. The passage A, in conjunction with the narrow annular ports, provides for double ad-

mission, as does the corresponding passage of the Allen valve; and like this it should be so placed that it never communicates with the exhaust, in order to save the residual steam it contains and to avoid condensation. The admission-edge and the exhaust-edge of a piston-valve extend all around its circumference, and for this reason a comparatively small diameter will give sufficient port-opening. Double-admission piston-valves are seldom provided with adjustable packing-rings, while it is common on single-admission valves. All piston-valves work in removable bushings, in which the ports are cut out. Judging from recent practice the double-admission piston-valve may become obsolete.

VALVES OF THE "IDEAL ENGINE."

Piston-valves of the "Ideal Engine" are shown in Figs. 23 and 24. One is a plain valve without rings, and the other is an "expansive" valve. Only one end of the latter is shown, and the scale is sufficiently large to show details of construction plainly.

The two rings, A, are made to fit the bore accurately, and

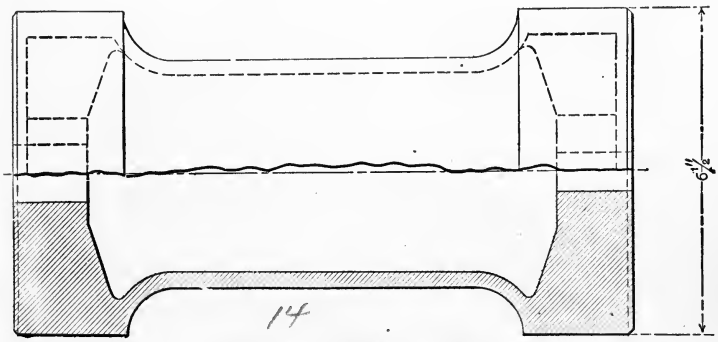


Fig. 23. "Ideal" Valve.

are then split to allow of some expansion. They are not "spring-rings," for they are thick and not made larger than the valve-bore of the cylinder. When the valve becomes leaky from wear the rings may be expanded a little by turning the head B. This head

has a ring-formed extension, on which are four cams or eccentric planes. These are accurately machined, and bear on four shoes C, which fit against the inside periphery of the expansion rings. Thus by turning the head the four shoes are simultaneously forced against the rings, and by thus expanding them, a delicate adjustment to the proper diameter is made. After adjustment is made, the nut D is screwed up tight, clamping the head and

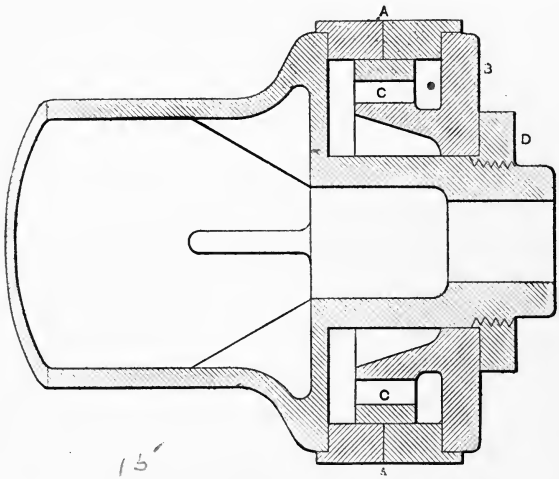


Fig. 24. The "Ideal" Expansive-Valve.

rings securely. The object is evidently to avoid the use of spring rings, which would wear the valve-seat, and would have to be very thin on small valves, and would be liable to break.

The builders of the Ideal Engine also use flat-balanced valves of the Straight-Line type; and in view of the fact that there has at times been a lively controversy about the relative merits of these two valve types, the following letter from the Ideal Engine builders is interesting:

DEAR SIR, — In reply to your letter of the 2d inst. Although we offer purchasers the option of taking the plain solid valve or our patented adjustable valve, three-fourths of our sales are for the plain solid valve. The valve-seats or

bushings, after being pressed into the steam-chest chambers, are bored out with a portable bar, and then a reamer put through, and a valve is fitted which has been previously ground on a Brown & Sharpe grinding machine for an accurate and close fit. By this method we get a valve that is perfectly balanced, and is practically steam tight, and requires less lubrication than valves of the flat type, and, consequently, gives excellent service. Three to five years' service can be had before the valve is worn sufficient to cause excessive loss of steam and require renewal; and then renewal is so easily and cheaply made by means again of the portable boring rig. The valves are light and give satisfaction in everybody's hands. The adjustable valve is all right, but in hands of careless or incompetent engineers it can be expanded so tight as to cause stripping of the valve gear.

We are also builders of flat balanced valves, but our greatest sales are the piston valve; and after having had experience with both types we believe the piston valve to be the best commercial design.

JAN. 6, 1902.

THE WESTINGHOUSE STANDARD VALVE.

Fig. 25 (on the following page) represents a sectional view of cylinders and valve of a Westinghouse Standard Engine. This is a two-cylinder vertical engine, and the cylinders take steam on the down-stroke only. Admission and exhaust are controlled by a single piston-valve, located between the cylinders. The pistons move in opposite directions; and the cut shows the left-hand piston (in sections) at its highest position, and the valve just commencing to uncover the steam-port; while the port on the other side is already uncovered sufficiently for a free exhaust. The live steam is confined in the annular space, surrounding the valve, and the hollow valve body and the upper and lower parts of the valve-chamber receive the exhaust steam. It will be seen that the valve arrangement is practically the same as that of a single-cylinder double-acting engine; the only difference is, that in the case under review, the steam-ports are on opposite sides of the valve.

The small piston above the valve is there to balance the reciprocating force of the valve and its connections, that is, to overcome the momentum of these parts. The exhaust steam and air confined in the upper part of the steam-chest above the

piston, is alternately compressed and expanded, and offers an elastic resistance, similar to that of a spring, alternately compressed and extended; and this elastic force is directly opposed to and partly neutralizes the reciprocating force of the valve.

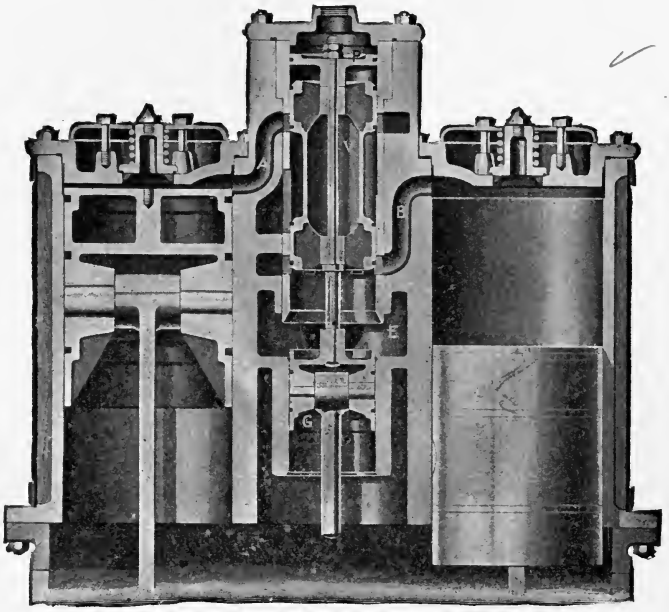


Fig. 25. The Westinghouse Standard.

The valve is driven by means of a shiftable eccentric controlled by a shaft governor, and the point of cut-off is automatically varied by changing the throw and angular advance of the eccentric; that is, by moving it across the shaft in a suitable manner.

VALVES ON COMPOUND ENGINES.

The valves of tandem-compound engines are frequently driven by a single shifting eccentric, which automatically changes the cut-off in both cylinders. In such case the high- and low-pressure cut-off must nearly coincide; for, the angular advance

being the same for both valves, the cut-off can only be made different by varying the laps; and as this changes the lead it is only permissible within narrow limits. Such arrangement is probably favorable to close regulation, and it has the advantage of great simplicity; but whether it has any economical advantage or disadvantage has not been practically demonstrated. If both valves have the same travel, the port-opening to the low-pressure cylinder will be very small comparatively, even if the length of the port is proportioned to the cylinder-bore; and on account of the high receiver pressure by early cut-off, a "nice" high-pressure "card" cannot be expected when the engine is under-loaded.

Some compound engines have a single piston-valve, driven by a shifting eccentric or a link-gear. The cylinders are arranged side by side, with the valve in an intermediate or parallel position, and both cylinders and valve-chamber are made in one casting; and in this combination the piston-valve has probably reached the acme of fitness and simplicity. Fig. 26 may represent a valve of this type for a small vertical compound-engine having two cranks oppositely disposed. The central part of the valve bushing is surrounded by steam, which is admitted through an annular port to the annular valve-space, which connects with the high-pressure cylinder, as shown. The valve has just opened for steam to the upper end of the high-pressure cylinder, and the exhaust from the lower end is just entering the low-pressure cylinder, while the low-pressure exhaust is escaping from the upper end through the hollow valve to the lower

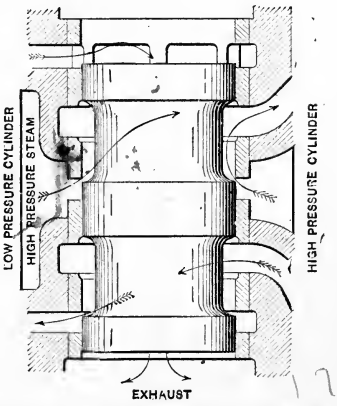


Fig. 26. Compound Piston-Valve.

exhaust-chamber. The steam-distribution is regulated by three ports: The central port admits and cuts off steam to the high-pressure cylinder, while the two other ports control the admission of high-pressure exhaust to the low-pressure cylinder, and also the exhaust from this cylinder. The exit from the high-pressure cylinder is not directly controlled by the valve; the expanded steam simply gets out as soon as it has a chance to go into the low-pressure cylinder.

Release and exhaust closure in the high-pressure cylinder become coincident with admission and cut-off in the low-pressure cylinder. The admission lap for the low-pressure side becomes the exhaust lap for the high-pressure side. This feature is open to criticism; for some lap is required to avoid excessive lead to the low-pressure cylinder, while for the high-pressure exhaust no lap or negative lap is preferable, for the compression of the intermediate exhaust may otherwise become excessive. Experience has established the fact that smooth running is hardly obtainable with excessive compression, but as to steam economy the only satisfactory answer can be furnished by a strictly scientific comparative test.

Considering that space which is *always* in direct communication with the cylinder is "clearance-space," it appears that there is no "receiver-space," for the low-pressure cylinder takes steam directly from the high-pressure clearance-space. This space is extraordinarily large, and when steam and exhaust are cut off late in the stroke it may cost something to fill it with steam at the commencement of each stroke; though under certain conditions this loss may be very insignificant, considering that this space never communicates with the atmosphere or the condenser; and it is probably very useful as a means of keeping down the compression in the high-pressure cylinder. These valves are usually fitted with packing-rings.

THE WESTINGHOUSE COMPOUND VALVE.

Fig. 27 shows the valve of a Westinghouse Compound Engine which is vertical and single-acting. The valve is of the piston type, hollow and light, and made steam-tight by means of four spring rings. It is operated by means of a bell-crank rocker from a shiftable eccentric, controlled by a shaft governor. The steam-passages are formed by a hard cast-iron bushing, forced into place. By this construction the ports can be machined to exact size and register. S and E are the steam and exhaust chambers around the bushing, into which connects respectively the steam and exhaust pipe. The valve is shown in its central position covering the high- and low-pressure ports. The ports consist of annular series of openings surrounding the valve. The high-pressure port is seen to the left, centrally over the high-pressure cylinder. Next to this is an open passage from the high-pressure cylinder to the space surrounding the neck of the valve. B is a by-pass valve, used only for starting the engine. Next to this is the low-pressure port, also covered by the valve.

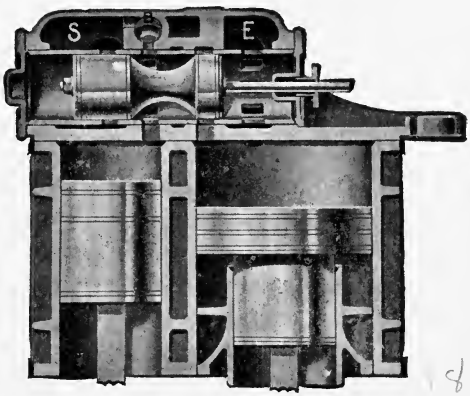


Fig. 27. The Westinghouse Compound.

The high-pressure port is only for admission to the high-pressure cylinder, while the low-pressure port is for admission and exhaust to and from the low-pressure cylinder, and it also controls the exhaust from the high-pressure cylinder. The space around the neck of the valve is part of the clearance space of the high-pressure cylinder; the pressure in this space

is always the same as that in the adjoining cylinder, being always in direct communication with it. It is never in direct communication with the low-pressure exhaust.

The large high-pressure clearance has been criticised from a theoretical standpoint, without regard to the fact that it cannot consistently be considered equivalent to, or compared with, the clearance of the low-pressure cylinder or that of a simple engine. This is what the Westinghouse people have to say to their critics: "We have no quarrel with theory, but only with mistaken interpretation of theory, and we simply submit that the practical results obtained from this design should set at naught the unsupported opinion of those who decry the method employed."

The pistons are of the trunk pattern, and it will be observed that the low-pressure piston has two diameters. The lower part works through an internal sleeve, or cylinder, which connects with the crank-case. By this construction the piston displacements in relation to the crank-case become equal, and suction and expulsion of air and oil through the vent-pipe in the crank-case is avoided. The air confined in the annular space underneath the large diameter, being alternately compressed and expanded, helps to overcome the inertia of the large piston.

THE VAUCLAIN VALVE.

The Vauclain Compound Locomotive has one high-pressure cylinder and one low-pressure cylinder on each side, and their volumetric ratio is about as one to three. They are cast in one piece with the valve chamber and saddle, the cylinders being in the same vertical plane, and close together. Fig. 28 shows the arrangement for "eight-wheel" passenger locomotives. The valve is located in the saddle of the cylinders casting, between the cylinders and the smoke-box. The steam-chest is bored out, and a bushing with accurately machined ports is forced in. The valve, shown in half-section in Fig. 29, is of

the piston type — double and hollow — and it controls the admission and exhaust of both cylinders. The pistons — secured to a common cross-head — move in unison ; and the exhaust steam from the high-pressure cylinder, becoming the supply steam for the low-pressure cylinder, is conveyed from the high-pressure port through the hollow valve to the low-pressure port at the opposite end. The operation of the valve can be readily understood by considering it as two valves combined — one for the high-pressure cylinder ports, and a shorter one for the low-pressure ports. The valve is shown in its central position ; and it will be seen that the laps are equal, and that steam is admitted over outside port edges and expelled over inside port edges. Live steam is conducted to both ends of the valve-chamber, and is finally exhausted in the central cavity, which connects with the exhaust-pipe.

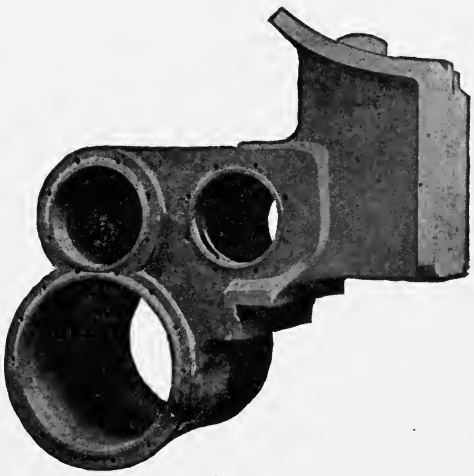


Fig. 28.

There is negative lap on the exhaust side, and the steam will be released in the high-pressure cylinder before it is admitted to the low-pressure cylinder. In the meantime it is stored in the hollow valve-body, which thus serves as a receiver, and makes it practicable to use admission lap for the low-pressure cylinder.

In locomotives it is necessary to restrict the exhaust outlet in order to get the benefit of expansion in the smoke-box, and the back-pressure tends to increase the compression in the low-pressure cylinder ; an early exhaust closure is, therefore, not desirable, and it is avoided by means of a negative exhaust lap.

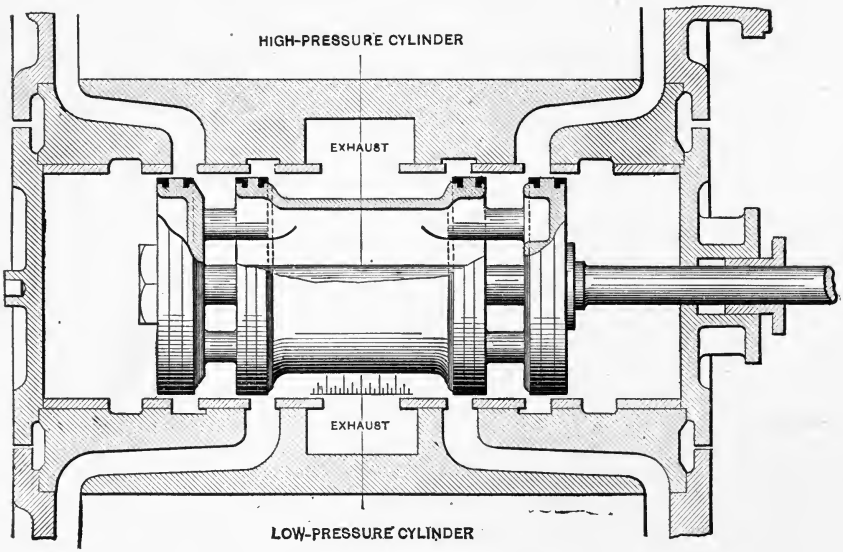
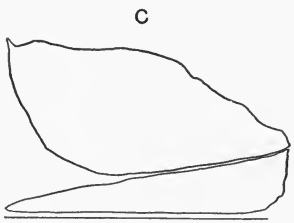


Fig. 29. The Vauclain Valve.

On the high-pressure side the exhaust lap is also negative, for otherwise the density of the receiver-steam would bring the compression up too high. It is not necessary to cut off earlier than half-stroke, for this gives six nominal expansions; and thus a larger port-opening and more decisive cut-off action is incidentally gained by compounding.



I.H.P. 1370
Fig. 30.

A reduced indicator diagram is shown in Fig. 30. The point of release is easily discernible in the high-pressure diagram, and it will be noticed that the expansion is checked before the low-pressure port is open. This is not remarkable, considering the small capacity of the receiver.

THE ALLFREE VALVE-GEAR.

There are several well-founded objections to the use of a single valve in variable cut-off engines and locomotives. The small port-opening, the slow cut-off action, the early release, and the extremely variable compression are not desirable features. There are various valve-gears in use, which give ample port-opening, but they do not correct the premature release and early exhaust closure. The ordinary valve-motion, when cutting off at half-stroke, is represented in diagram 31. As explained in chapter 1, the two lap lines must be parallel; and arcs 4–6 and 1–9 must, therefore, necessarily represent equal intervals of time, that is, the interval between cut-off and release, and between exhaust closure and admission, must, under all conditions, be equal. When the valve cuts off at half-stroke, the release and compression may be quite satisfactory; but at earlier cut-off, when the full benefit of expansion is obtained, the exhaust action becomes less satisfactory. The compression may not become excessive if there is considerable clearance; but the variableness of the compression is unavoidable; and it may safely be assumed that much clearance is not conducive to steam economy.

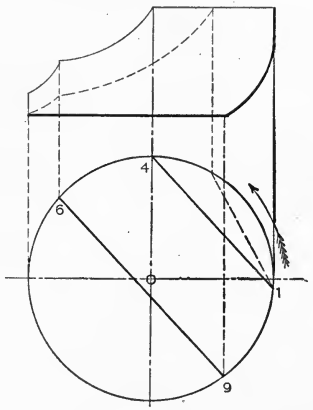


Fig. 31.

If the exhaust-cord would remain as shown in Fig. 31, a small clearance would suffice, and the release would be satisfactory. Practically, these are the conditions existing in the Allfree single-valve engine. The valve-gear is shown in Fig. 32. There are a toothed sector, a pinion, and a small eccentric mounted in the rocker-arm. While the rocker-arm is actuated

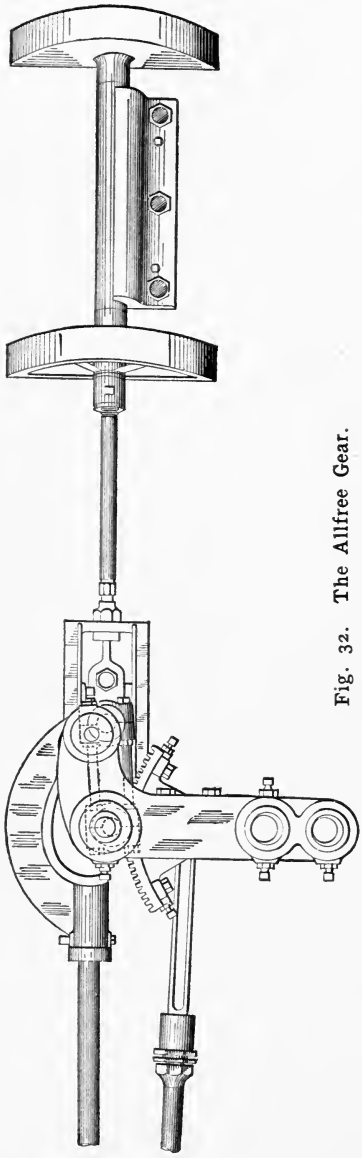


Fig. 32. The Allfree Gear.

in the ordinary manner from the governing eccentric, the sector is operated from a fixed eccentric keyed upon the shaft. By this means a rotary motion is imparted to the pinion-shaft. This shaft is made from a solid piece of machinery steel, having an eccentric of about $\frac{3}{8}''$ radius formed at one end. This eccentric is connected through a small link to the valve-stem. In operation, if the rocker arm is allowed to stand fixed, and the engine rolled over, the eccentric on the pinion-shaft would move the valve three-fourths of an inch. If the rocker arm is allowed to operate, the small eccentric will make nearly a whole turn while the rocker moves from one extremity of its throw to the other; and it will be clear that, during the rotation of the eccentric, and the movement of the rocker arm, at certain predetermined points the two movements will coincide and others will oppose. It gives the valve a high speed at the time of opening and closing, and a slow speed or pause during expansion. Thus the valve is caused to open for a given cut-off about twice as wide as in an

ordinary automatic. The advantage thus secured for the steam admission is likewise secured for the exhaust.

In Fig. 33 let the main eccentric be held by the governor in position B for latest cut-off, and let E mark the position of the fixed eccentric, then angle V represents the angular advance of the main eccentric in relation to the fixed eccentric. When the main eccentric has been moved to position B′ for shortest cut-off the angle between the eccentrics has been increased to 180°. The throw of the main rocker-arm is determined by the throw of the main eccentric, and the rotation of the small rocker-eccentric is determined by the relative position of the two eccentrics on the shaft. Therefore, in proportion as the main eccentric is advanced toward the crank the motion of the rocker-eccentric becomes later relative to the rocker motion. Now the rocker-eccentric may be placed in any desired position in relation to the rocker, and it may be so put that when the valve is cutting off at half-stroke release and exhaust closure will be timed just as with an ordinary valve-gear; but when the governor eccentric is moved forward the rocker-eccentric will lag behind to the same extent, and the whole series of alternatingly increased and diminished valve movements will occur a little later in the stroke. The effect of this will be to lengthen the interval between cut-off and release, and to shorten it between exhaust closure and admission; and this is just what is needed in order to get a later release and less compression, and thus the ideal conditions represented in Fig. 31 may be nearly realized.

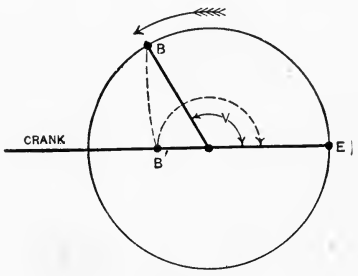


Fig. 33.

Fig. 34 represents an indicator diagram from the Allfree engine. It will be recognized as a remarkable single-valve diagram, and further comment is unnecessary.

The valve, Fig. 32, is in the shape of two circular segments; it is extremely light, and is balanced between an arched hood and the valve-seat. Steam is admitted to the inside space, and it partly surrounds the cylinder. The large port-opening obtainable with the Allfree valve-gear obviates the necessity of complementary ports, and the steam-passages being direct and very short, the clearance is reduced to a minimum. It will be ob-

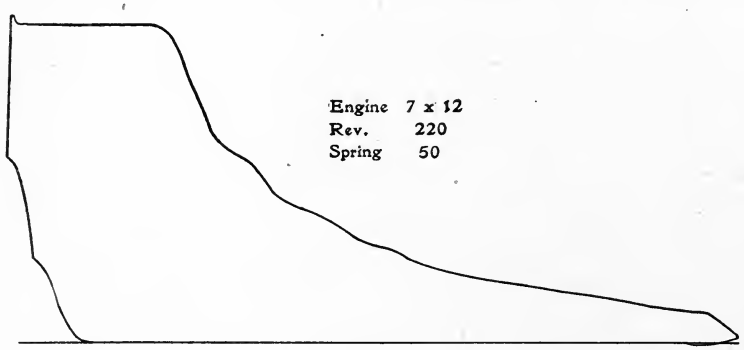


Fig. 34.

served in Fig. 32 that the weight of the valve is carried by a long adjustable bracket bearing, which keeps it from bearing on the hood.

This valve-gear may be used on any single-valve engine whether steam is admitted inside the valve between the ports or on the outside; in either case the fixed eccentric will be placed in line with the crank and on the opposite side of the main eccentric.

CHAPTER III.

FOUR-VALVE SYSTEMS.

INTRODUCTORY REMARKS.

FOUR separate valves are often used, two at each end of the cylinder, one for steam and one for exhaust. If all four valves were positively driven by one eccentric the steam distribution would be exactly like that of a common D-valve — it would, in effect, be the D-valve dissolved in its four component parts. But the four-valve system has points of superiority not possessed by the single combination valve.

1. Steam and exhaust passages become short and direct.

2. The valves are not balanced; but the unbalanced pressure will only be that due to the difference between the pressure in the cylinder and that in the adjoining steam and exhaust chambers, and the full effect of this will be intermittent, and on the port area only.

3. The separation of the valves makes it possible to reduce the valve motion by means of toggle motion of the valve rods. The port must be covered during one-half the period of revolution at least, and if the motion of the valve is the same in opposite directions from a central position, as it must be with the D-valve, the greater part of the motion must generally occur after the port is closed. This can be avoided when the valves are separate, for then each valve may be separately attached to a special rocker or wrist-plate in a manner to produce a reduced valve-motion during the closed period.

CORLISS VALVES.

Fig. 35 represents diagrammatically the Corliss valves and wrist-plate connections for one end of the steam cylinder. It shows how the toggle motion of the radius-rods operates to shorten the movement of the valve after the port is closed. As the angular motion of the valve is reduced by the toggle action, it becomes possible to use smaller valves than otherwise practicable, because a given angular motion yields a larger port-opening than could be had with a straight motion of the rod.

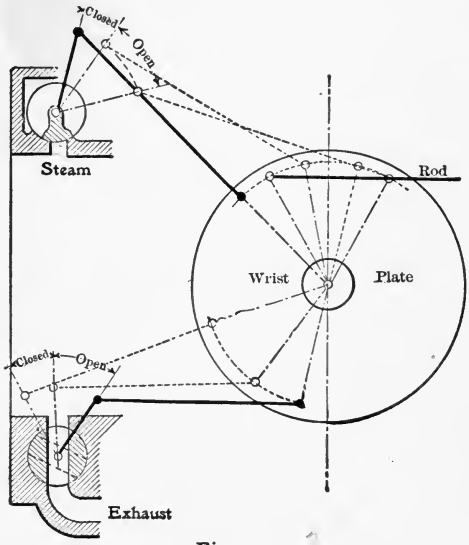
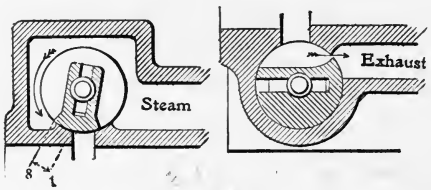


Fig. 35.

Figs. 36 and 37 show the construction of the first Corliss valves. The valve stem has a flat extension which fits the whole length of the valve, and has a journal at each end.

The Corliss steam-valve, shown in Fig. 38, is a simple circular slide suitably guided. The slide and cylindrical parts at each end are now often cast in one piece, and the valve-stem terminates in a T-head which fits in a slot across the end of the cylindrical part.



Figs. 36 and 37.

These valves are often made double-ported to avoid the

necessity of using large diameters or to reduce the motion. The objection to double ports is the increased clearance space they make. The clearance may be reduced to a minimum by putting the valves in the cylinder heads, and this would, no doubt, become the general practice if it did not involve somewhat complicated steam connections.

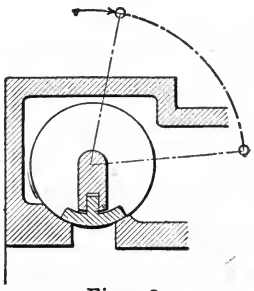


Fig. 38.

Double-ported steam and exhaust valves are shown in Fig. 39. Fig. 40 represents three-ported valves for the lower head of a 60-inch low-pressure cylinder of a cross-compound vertical engine, designed by Messrs. Rice & Sargent. The steam-valve is held up to its seat by end-

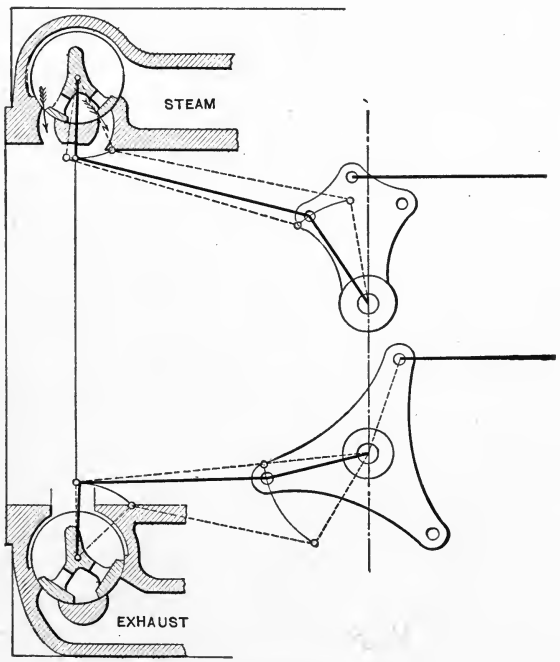


Fig. 39.

gibs, supported by flat steel springs, and these are adjustable by means of screws from the outside.

The exhaust-valve is a hollow shell with closed ends. It is separated from the bottom of the cylinder by a loose ported valve-seat plate, the upper side of which is flush with the cylinder head. This plate is held freely between planed jaws in the cylinder head; and the steam pressure in the cylinder keeps it against the valve, and prevents the steam from passing inside the valve when the ports are closed. The opposite side of the valve is provided with a single wide port, and is always open to the exhaust-chest. This construction gives less clearance than the ordinary Corliss exhaust valve.

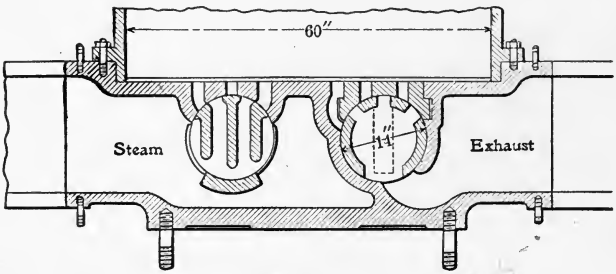


Fig. 40.

The operation of the Corliss induction valve is quite different from that of a common slide-valve. The special feature is a disconnecting device, which allows the valve to be closed by a spring or vacuum dash-pot, or other independent agency. The eccentric moves the valve till it is released by the knock-off cam, and it is then closed by the dash-pot. The valve is perfectly at rest during the expansion period and the greater part of the exhaust period, but during the exhaust period it is caught by the "steam-hook," and slowly started, so as to admit steam near the beginning of the forward stroke. The amount of opening and the period of admission depend on the position of the knock-off cam, and this is in turn determined by the speed-

regulator. When the steam-hook arm strikes the knock-off cam the hook is released, and the valve is closed by the dash-pot. The result is a nearly ideal valve-motion.

LIMITATIONS OF THE CORLISS GEAR.

When a single eccentric drives both steam and exhaust valves the range of cut-off is limited to about half the piston-stroke. This will become obvious by considering the following necessary conditions:

1. After the eccentric has reached the extreme of its throw in either direction all valve-gear motions are reversed.

2. The steam-valve must be released before the eccentric-motion is reversed, for if the hook does not strike the knock-off cam during the forward motion it cannot strike it during its return motion.

3. The maximum exhaust opening, or the middle of the exhaust period, must occur when the eccentric is at the end of its throw.

Now, in order to get release of the expanded steam in the cylinder before the commencement of the return-stroke and exhaust-closure a little before the end of the return-stroke, the middle of the exhaust period, or the extreme throw of the eccentric must evidently occur before the middle of the return-stroke; and therefore the extreme throw of the eccentric in the opposite direction must occur before the middle of the forward stroke, and the valve must be released before that point is reached, if released at all.

The point where steam is cut off is not coincident with the action of the cut-off cam, for it takes some time to close the valve. With a long piston-stroke the closing period is comparatively short, but it is far from being "instantaneous," as seems to be the prevalent impression. The return-motion of the valve and dash-pot starts comparatively slow, as indeed all motion does; but this slow motion is not observable on account of the short-

ness of the interval, and by overlooking this fact a false conception of the dash-pot action is formed.

A good vacuum dash-pot closes the valve in about one-sixteenth of a second at late cut-off, or during one-twelfth of a revolution if the engine makes eighty revolutions per minute; and the closing motion will in that case cover about one-sixth of the stroke. With short stroke and high piston-speed the latest cut-off would be much later than mid-stroke.

The releasing gear is seldom used on high-speed engines, because the sudden and frequent action will cause it to wear quickly and become unreliable; and, as a sharp cut-off is not attainable at high speed, there is not sufficient justification for the use of a somewhat complex and delicate gear. One hundred revolutions per minute may be assumed as a practical limit for the Corliss gear.

This speed-limit of the Corliss gear is of no consequence in connection with large engines, where great piston-speed may be obtained with less than one hundred revolutions per minute; and, for obvious reasons, in such cases simplicity or fewness of parts is a consideration of less moment.

It will be understood from the foregoing that late release and late exhaust closure are conditions imposed by the single-eccentric-valve gear; and these conditions agree very well with moderate piston speed, but at higher speed earlier release and more compression may be required.

This may be effected by moving the eccentric forward on the shaft; but the reversing of the steam-hook motion would then also occur at an earlier stage of the forward stroke, and the range of cut-off would be correspondingly shortened.

Earlier exhaust closure could be had by giving the exhaust-valve more lap, but this would involve a later release of the expanded steam at the end of the forward stroke. On the other hand, shortening the exhaust-lap would give earlier release, but insufficient or no compression.

THE SINGLE-ECCENTRIC VALVE DIAGRAM.

Let the circle, Fig. 41, represent the path of the center of the eccentric turning in the direction shown by the arrow, and let the horizontal diameter represent the throw of the eccentric. When the center of the eccentric is at point 1 the steam-port opens, and if the valve is not released before the end of the throw at point 3, the port will close again at point 4. The exhaust-port opens at point 6, and is closed at point 9. The crank is supposed to be at its dead center C when the eccentric is at point 2, just after opening the steam-port. When the crank is on the opposite dead center, the eccentric must be at point 7, diametrically opposite point 2; and the point of release at 6 occurs a little before this, as it should do. Arc 1–2 represents period of steam-lead, and arc 6–7 represents period of exhaust-lead, and arc 9–1 represents the compression period. When the eccentric is at point 3, the valve-gear motion is reversed, and the crank has not yet arrived at point 10; the valve is, therefore, necessarily released before mid-stroke, if it is released at all. Note that the chords must be parallel if there is only one eccentric, and that in order to get release before the end of the stroke, the diameter 2–7 must cross the chord 6–9. If, in order to extend the cut-off, point 2 be moved back to point 10, the diameter 10–5 could not be made to cross the exhaust chord in any position, and release would then take place after commencement of the return stroke. This simple diagram fully illustrates the limitations of the single-eccentric valve-gear; but it may be profitable to turn the figure, and let diameter 2–7

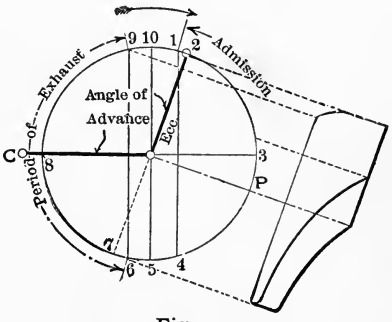


Fig. 41.

represent stroke of the piston, and let the circle represent the path of the crank. Points 1, 2, 3, . . . will then mark significant positions of the valve-gear in proper **relation to the piston-stroke,** and an imaginary indicator-diagram may be constructed, as shown. Note that by this arrangement the position of the eccentric relative to the crank is not considered; but it clearly shows the result when the eccentric is moved forward on the shaft, in which case the whole cycle of events occurs earlier in the stroke; that is, the lead is increased, the range of the cut-off is diminished, and there will be earlier release and more compression.

The steam-hook connects with the valve at or near the extreme of its throw; and the eccentric-motion 8–1 determines, therefore, to what extent the valve must overlap the steam-edge of the port when at rest. Note that during this period the valve should move as little as possible (8–1 Fig. 36), and the radius-rod must therefore have considerable lateral motion before the valve opens.

When the wrist-plate is in its central position the eccentric is at point 10, and the lap of the steam-valve in this position is determined by the eccentric-motion 10–1. The corresponding exhaust-lap is determined by the eccentric-motion 5–6. The exhaust-chord 9–6 could evidently be moved to the other side of the center, in which case the exhaust-valve would not cover the port at half-throw; or the chord could be placed centrally, which would make the exhaust-period cover exactly one-half revolution, and would put the exhaust-edges "line and line" in the half-throw position; but such changes would give earlier release and less compression. Lap or clearance can be laid out very accurately on the drawing-board, and it should be marked on the drawing. The great adjustability of the Corliss gear makes it easy to set the lap by some rule of thumb after the engine is erected, and a little variation in the points of release and compression may not be of much practical consequence; but when a "good" indicator-

card is required there may later be some tedious resetting to do. There should always be marks at the end of the valve, inside the rear-bonnet, in line with the steam-edges of valve and valve-seat.

At point 3 the motion of the steam-hook is reversed; and if arc 3–P represents the period of dash-pot movement, arc 2–P will represent the admission-period for latest cut-off, as determined by the knock-off cam, which, in this case, would be one-quarter of a whole revolution or half of the piston-stroke; but if under heavy load, the speed-regulator puts the knock-off cam out of reach of the steam-hook, the engine will take steam during period 1–4, or nearly full stroke.

TWO ECCENTRICS.

In order to obtain a greater range of cut-off in Corliss engines, a separate steam-eccentric is used and has become quite common. With two eccentrics the admission and exhaust-periods can be adjusted independently, and steam may be cut-off anywhere, nearly to the end of the stroke.

In order to start the valve without shock, the hook must connect with it when near the end of its throw; and the steam-eccentric may be so placed in relation to the crank, that this connection is made near the end of the return piston-stroke, for the valve need not overlap the port more than from one-half to three-fourths of an inch when closed.

The arrangement of steam-rods exemplified in Fig. 35, is in every respect satisfactory in connection with a single-eccentric valve-gear; for in that case a slow initial valve-motion is imperative, and it is obtained by the lateral movement of the radius-rod. But with two eccentrics a quicker initial motion is feasible and desirable, and it may be obtained by reversing the valve-motion and admitting steam over the top of the valve (Fig. 39).

Separate eccentrics require separate wrist-plates, and their

centers may be located one above the other, as shown in Fig. 39.

Figs. 42 and 43 show how the eccentrics may be placed on the shaft. The steam-eccentric is at point 2, Fig. 42, and the exhaust-eccentric is at point E, Fig. 43, and the crank is at its dead center at C. Individual eccentric circles are shown for the sake of clearness, and the notation is the same as in Fig. 41. An imaginary motion of the eccentric-arm will point out the various events. Referring to Fig. 42, near point 8, at the extreme of the throw, the hook connects with the steam-valve, and at point 1 the steam edges are on the point of separating, and the eccentric-motion, 8–1, determines, therefore, the initial

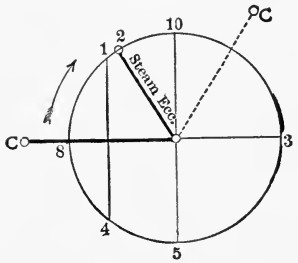


Fig. 42.

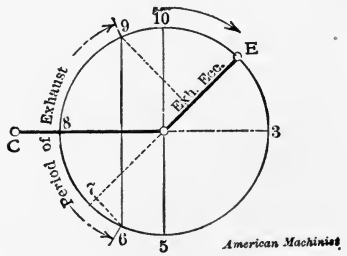


Fig. 43.

valve-motion. When the eccentric is at point 2 the crank is at its dead center, as shown. At point 10 the steam-wrist-plate is in its central position, and in that position the valve does not cover the port, as with the single-eccentric gear; but the port is open to a certain extent, determined by the eccentric-motion 1–10. Point 3 marks the extreme of the throw, and the corresponding position of the crank is at C′, at about three-quarters of the piston-stroke, and the limit of automatic cut-off is a little later. If the hook does not strike the knock-off cam the valve will remain open till closed by the return stroke of the eccentric at point 4, near the middle of the return piston-stroke.

The exhaust action is discernible from Fig. 43. It is similar to the single-eccentric action, but with this difference, that

the release, at point 6, occurs at about 90 per cent of the piston-stroke, and the exhaust is cut off at about 70 per cent of the return-stroke, at point 9. The motion of the exhaust-valve, after it has closed the port, is determined by the eccentric-motion 9–3–6, and full port-opening is obtained by the eccentric-motion 6–8. The motion 9–10 determines the exhaust-lap when the wrist-plate is in its central position.

A valve-gear designed to be operated by a single-eccentric cannot very well be made to cut off much later than at half-stroke, even when a separate exhaust eccentric is added. For the slow initial valve-motion requires at least half the throw of the eccentric, and the other half is not sufficient for a late cut-off; and it will readily be seen from an inspection of Figs. 35 and 39 that a quicker initial valve-motion in Fig. 35 would involve radical changes in the valve-gear. However, the range of cut-off may be increased some by moving the eccentric back, sacrificing the lead; and to this there is no objection when it does not involve later release.

The advantage gained by a second eccentric would consist in more compression and earlier release.

CORLISS VALVE DIMENSIONS.

The steam- and exhaust-ports of Corliss engines are usually made as long as the diameter of the cylinder-bore, and steam- and exhaust-valves are usually of equal diameters, but these diameters do not vary as the bore of the cylinders, as might be anticipated. The proportion varies from one-third of the bore in small engines to one-sixth for large low-pressure cylinders. The port is not necessarily proportioned according to the diameter of the valve, but the port-opening will be nearly so. It is considered a safe rule to make the width of port sufficient to allow a mean velocity of 8000 feet per minute of the entering steam, and 6000 feet per minute of the exhaust; but a smaller port area may consistently be allowed for large cylin-

ders. The port-opening is in proportion to the travel of the valve, which is practically limited by the diameter.

DIRECTIONS FOR SETTING THE VALVE-GEAR.

Adjust length of eccentric-rods to give wrist-plate equal travel on both sides of center-mark on bracket. Adjust length of radius-rods to give proper lap with wrist-plate in its central position. Move wrist-plate to end of its travel either way (as marked on the bracket), and adjust length of drop-rods to let the hooks freely engage the catch-blocks. Put crank on dead centers, and set eccentric ahead of the crank, sufficiently to give the proper lead. Raise governor to highest working position, and adjust length of rods so that the knock-off cams will just keep the hooks off the catch-blocks; or some initial motion may be allowed, but not enough to open the port.

VALVES OF THE PORTER-ALLEN ENGINE.

An interesting instance of the four-valve system is found in the Porter-Allen engine. There are two steam-valves on one side of the cylinder, and two exhaust-valves on the other side, and the valves move in a direction parallel with the cylinder-bore. The valves are flat; and the steam-pressure on opposite valve-faces is absolutely balanced by means of pressure-plates which fit closely against the back of the valve, and the steam-passages to and from the cylinder are short and direct. The valves are driven by an Allen link, and the position of the link-block is automatically changed by a Porter governor. The period of admission is changed by varying the travel of the valve, while the lead remains constant; and the result is the same as what may be accomplished by a shifting eccentric and a shaft governor. The exhaust-valves are driven from a fixed point on the link, and they have an invariable motion, precisely as the motion derived directly from an eccentric fixed on the engine-shaft. The steam valves have separate valve-stems; and

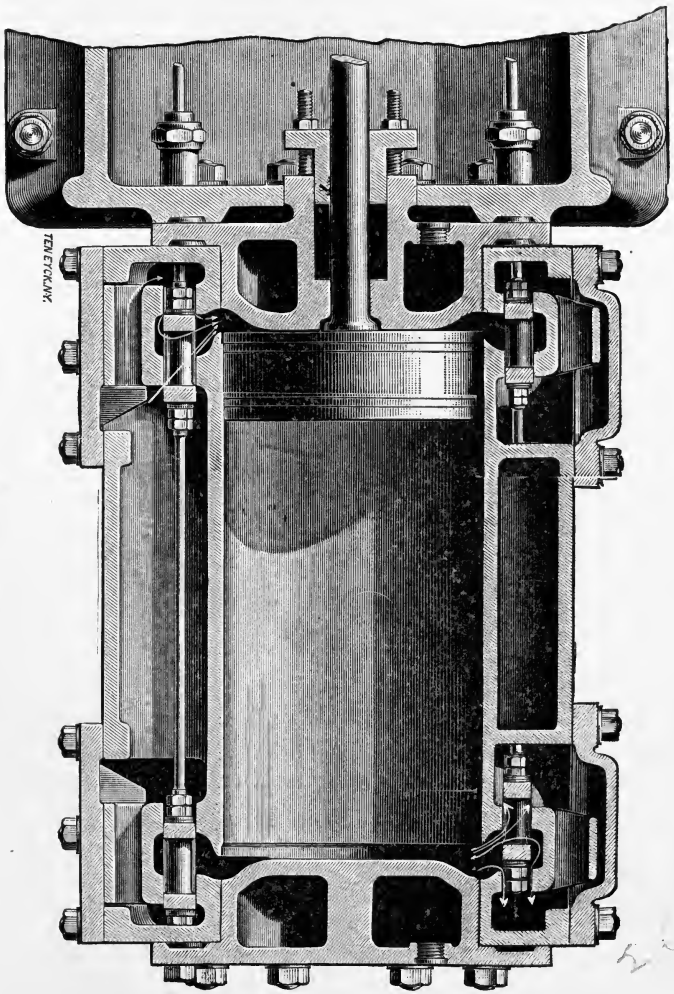


Fig. 44. The Allen Valves.

they receive their motion from two bell-crank levers, which, like the Corliss wrist motion, greatly reduce the movement of the valve after it has closed the port.

Fig. 44 shows a horizontal section through cylinder and valves; and it shows the course of the steam through four open-

ings into the steam-port, and also from the exhaust-port into the exhaust-chest. The pressure-plates are arched so as to make them practically unyielding to steam-pressure, and there is left a free passage for steam through the opening of the arch. The pressure-plates of the admission-valves are adjustable to more or less close contact with the valve by means of short inclines, against which they are held by the steam-pressure in the steam-chest. They are adjusted by a short lateral displacement on said inclines by means of screws, which extend through the bottom of the steam-chest. The pressure-plates of the exhaust-valves are bolted to their seats. The valves are formed like open rectangular frames; and the opening in the frame is made wide enough to reach the edge of the valve-seat when the port opens, otherwise there would not be four simultaneous port-openings. The space inclosed by the valve-frame adds to the cylinder clearance; and as the valve is perfectly balanced the main object of the bell-crank levers is, apparently, to shorten the valve and limit the clearance space. As the steam-valves do not move in unison, each valve must have its individual valve-stem, the stem of the rear valve passing through the front valve. The exhaust-valves have no lap; that is, in its central position the exterior edge of port and valve are "line and line." The valve, therefore, opens and closes quickly, and only a short travel is required. It is obtained by means of a double-ended reducing-rocker, from the short arm of which motion is imparted directly to both exhaust-valves.

Fig. 45 shows one of the exhaust-valves used on the first Porter-Allen engine. There is no pressure-plate; but an open frame, of nearly same size as the valve-frame, is held against the back of the valve, as shown. This frame is bolted to a copper diaphragm, which is clamped to the steam-chest cover; and the steam from the cylinder, having access to the back of this diaphragm, keeps the frame against the valve and holds it to its seat. During the exhaust period this valve is absolutely bal-

anced; and the pressure during the admission period, when a tight joint is needed, is not excessive. There is a projection on the chest-cover, which nearly fills the space inside the valve-frame, and thus the clearance was reduced as much as possible.

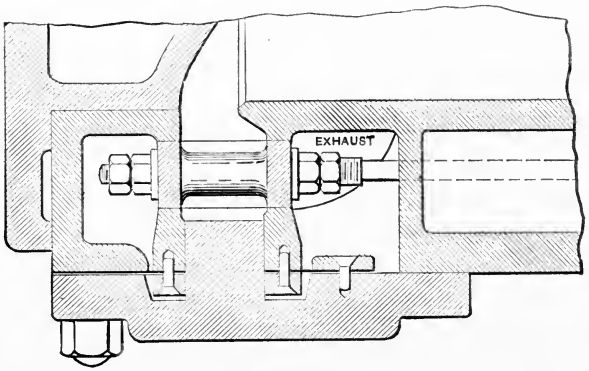


Fig. 45.

The valves and valve-gear of this engine are the invention of John F. Allen, and in the history of engineering will hold a prominent position, not only on account of their originality and excellency, but because they first made high speed in a variable cut-off engine practicable.

GRIDIRON VALVES.

Flat multiported, or "gridiron" valves, Fig. 47, are often employed as a means of obtaining sufficient port-opening by short-valve travel, and they have lately been extensively used on four-valve engines. Flat valves are better adapted for multiporting than Corliss valves. They give straight steam-passages, and they can readily be fitted with removable seats of hard, close-grained iron; which is of some importance for high-duty engines under high steam-pressure, and the accurate spacing and finishing of the ports is more readily obtainable than with

valve-seats cast on the cylinder. These valves have also considerable bearing surface, equally distributed, and they can be forced from their seat when much water is carried into the cylinder. Some offset to these advantages is found in the fact that the bars of the exhaust-valve and the steam-valve-seat expose considerable surface to the entering steam. The short travel of multiported valves is an advantage, if not a necessity. It is also a matter of convenience, for it provides for a valve operating mechanism of practical construction and of moderate proportions.

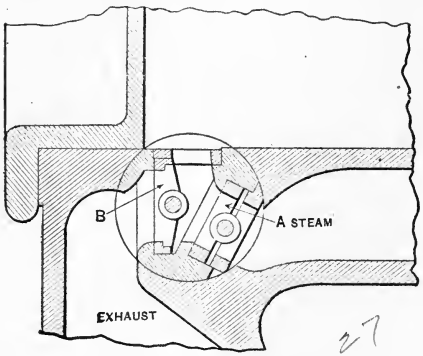


Fig. 46. The Hill Valves.

THE HILL VALVES.

A unique combination of gridiron valves, designed by Mr. Edward K. Hill, is represented in Figs. 46 and 47. The main

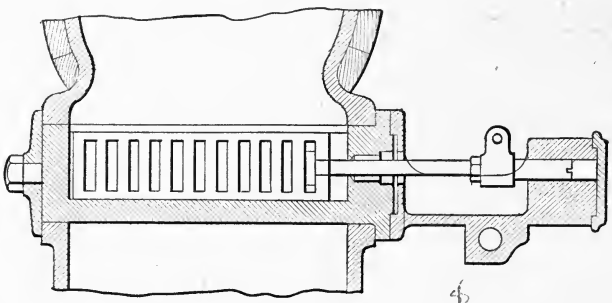


Fig. 47.

characteristics of this system is apparent from Fig. 46, which represents a cross-section through a pair of valves and valve-seats. A is the inlet-valve, which controls the admission and

cut-off, and B is the exhaust-valve. Both valves are seated in a round, open cast-iron plug, of a hard, close-grained mixture, of which Fig. 47 is a longitudinal section, exposing the face of the exhaust-valve. The plug containing the valves is pushed in endwise, and is kept in place by a single nut, as shown; and it may be removed and replaced very quickly — a feature which practical enginemen will appreciate. The valve-moving mechanism (not shown) is attached to the front end of the plug, and contains a pair of toggle-levers, by which the exhaust-valve is moved endwise, and which leave it nearly stationary between the exhaust periods. The inlet-valve is worked by an oscillating pusher, which engages an adjustable tappet on the valve-stem. The pusher is below the valve-stem; and at a certain point of its forward and downward movement it leaves the tappet, and the valve, being released, is quickly closed by a helical spring assisted by the steam-pressure on the valve-stem. The point of cut-off is changed by vertical adjustment of the tappet, and which is accomplished automatically by a concave lifting cam, operated by the governor. To prevent concussion by the closing of the valve, there is a dash-pot in the head end of the bracket, at the end of the valve-stem. It will be observed that this combination embodies the fundamental principles of the Corliss-gear, though there is no apparent similarity in the construction. The valves — one pair at each end of the cylinder — are driven from eccentrics on a revolving shaft, which runs alongside the cylinder, and which also drives the governor. There is an eccentric for each valve; and the range of cut-off is, therefore, not restricted in any degree, and the adjustability for lead, release, and compression is practically unlimited. The peculiar releasing mechanism and the short travel make a speed of 150 revolutions per minute practicable.

The peculiar construction of these valves permits of both valves being placed directly under the cylinder, which makes less clearance than with two separate steam-passages at each

end of the cylinder; and this location of valves affords a natural drainage from the cylinder; and the steam inlet, acting somewhat like a separator, will serve to precipitate the water. Being stowed between the valve-seats, it is left in position to be swept away by the exhaust without entering the cylinder. It can hardly be disputed that the proper place for the valves is directly below the cylinder-bore; and it is, no doubt, practical difficulties which have prevented such disposition of the valves from becoming more general. The system here described apparently solves the problem without a drawback; for the valves are readily accessible without any bolted joints, and there is no interference with steam- and exhaust-passages.

According to the builders, this system is the outgrowth of the Wheelock system, which it has now entirely superseded; and a comparison of the two systems becomes, therefore of great interest.

THE WHEELOCK VALVES.

Originally these valves were cylindrical, and semi-rotative like the Corliss valves; but there is a rational, functional difference. In the Wheelock system one valve serves for admission and release, and a smaller valve, in close proximity to this, cuts off the steam at a point determined by the governor; and in that respect the system differs from all others described so far, and it belongs properly in the independent cut-off system, fully discussed in the following chapter. These valves were supported by hardened steel trunnions in hard bushings, which partly relieved the pressure on the valve-seat; and the valve being tapered,

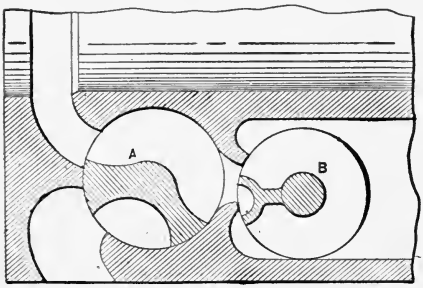


Fig. 48. The Wheelock.

the working faces were brought into close proximity by proper end-adjustment. Under moderate steam-pressure the contact will be light; but, obviously, this mode of balancing is not satisfactory under very high pressure, for the valve being supported at the ends only, the deflection in the center will cause it to bear hard on the valve-faces; and as the valve-barrel cannot readily be lined, the wear of the central portion may become quite considerable.

Fig. 48 represents a section through the valves. A is the main valve, and B is the cut-off valve or wiper. It will be noticed that the main valve has a large exhaust-cavity, and is really a cylindrically formed D-valve. The exhaust-cavity greatly increases the unbalanced area, and it weakens the valve-body; and the travel becomes considerable compared with the width of the bearing-surfaces. The cut-off valve has a cavity for double admission. It is worked from the main-valve lever by means of a Corliss crab-jaw, and it is closed by a tripping-mechanism under control of the governor, similar to that of the Corliss engine; and the limitations imposed by this gear are the same as those of the single-eccentric Corliss gear. This is probably the only engine which has an independent cut-off valve worked by a Corliss releasing-gear.

As the advantage of higher steam pressure became apparent, the inventor, Mr. Jerome Wheelock, changed his valves into gridiron valves, which he seated in skeleton plugs.

These valves, unlike the Hill valves, were actuated sidewise by an inside rocker-shaft and cranks.

CHAPTER IV.

INDEPENDENT CUT-OFF.

INTRODUCTORY REMARKS.

It has been explained in Chapter I. how the exhaust may be satisfactorily regulated by separating the exhaust-valve from the admission-valve. Better results can be obtained by having two valves, one of which controls the admission and exhaust, while the other controls the cut-off only; for by such arrangement larger port-opening and sharper cut-off is obtainable, while the points of lead release and compression are fixed. The usefulness of this arrangement is somewhat limited, however, by the fact that ample compression and a wide range of cut-off cannot exist at the same time; but this limitation would only be of consequence in "high-speed" engines having considerable "clearance."

Cut-off valves are slide-valves whose only function is to cut off the connection between the steam in the cylinder and that in the steam-chest before the piston has completed its stroke.

In England, and on the Continent, they are named Expansion Valves, because, by cutting off the cylinder-steam from the main steam supply, they provide for expansion of the steam in the cylinder. When these valves came into use there were no single-valve automatic cut-off engines made, and effective expansion was obtained by expansion-valves only.

The cut-off valve is auxiliary to the main valve; it opens and closes a port through which the steam must pass before it enters the main steam-port, or cylinder-port. The function of the main valve is to admit and exhaust the steam through

the main ports, and it determines the points of admission, release, and exhaust closure. The only absolute requirements of the cut-off valve are, that it shall open each of the auxiliary steam-ports not later than the commencement of each piston-stroke, that it shall close it again at a certain point in the stroke — which may be varied by hand, or automatically changed by the governor, — and lastly that it shall not open for steam again before the main valve has closed the cylinder-port.

THE CUT-OFF VALVE ON A STATIONARY VALVE-SEAT.

Fig. 49 shows the Gonzenbach cut-off. The cut-off valve slides on a ported partition above the main valve; and the steam, after passing into the lower steam-chest, is admitted through either one of the cylinder-ports which happens to be open. This cut-off valve is double-acting — it cuts off by both port edges, moving in opposite directions — for there must be one cut-off for each stroke of the engine, twice in the period of rotation. It is proper to consider this valve as a pair of valves operating on one port, each valve closing and opening the port in turn — this being, in fact, a distinctive feature of the Gonzenbach cut-off.

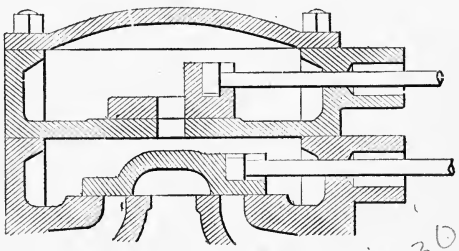


Fig. 49. The Gonzenbach Valve.

When there is a separate passage from the cut-off valve to each cylinder-port, steam is cut off by one of the port edges only. A slide of the simplest description, to cut off at one end of the cylinder, is shown in Fig. 50. It is shown in its central position; that is, when the eccentric is at half-throw. It has lap, and will, therefore, cover the port entirely during more than half the period of rotation. Fig. 51 represents a cut-off valve

with negative lap. It will cover the port during less than half the period of rotation. In both cases steam is cut off by edges A and B. The motion of these two valves is represented by diagrams 52 and 53 respectively. The second valve has only

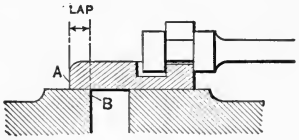


Fig. 50

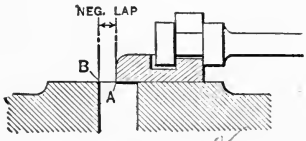


Fig. 51

half the throw of the first one, but it yields as large a port-opening and a sharper cut-off. It could not be used as a main admission-valve, because it is open during more than half the period of rotation, but it is the proper construction for a cut-off valve.

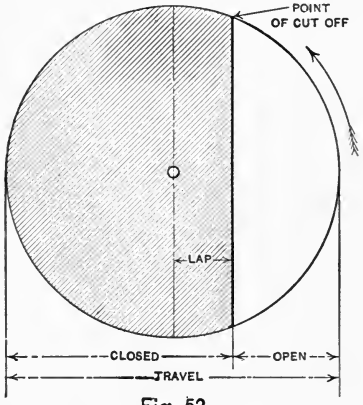


Fig. 52

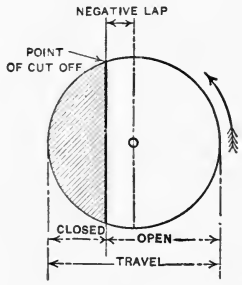


Fig. 53

The right-hand side, or half, of the valve of the Gonzenbach cut-off operates as represented in diagram 53, but the left-hand side, cutting off by the opposite port-edge, covers the port part of the time when it would be left open by the right-hand side of the valve, and *vice versa*. The motion diagram of this valve is represented in Fig. 54. Note that in this diagram the port-

opening is shown on both sides of the center line, and it should be measured from one of the two lap lines toward the center line. The point of cut-off may be automatically varied by rotation of the cut-off eccentric on the shaft; but in that case the cut-off is limited to one-half of the piston stroke, for the valve must be capable of holding the port open or closed in the same length of time, the limit of which, in this case, is one-fourth of the period of rotation. This valve is not used now, for other reasons besides its limited range of cut-off. The steam in the lower part of the steam-chest will expand with the steam in the cylinder, and under light loads the cut-off will be very short, and under these conditions the speed cannot be properly regulated by the operation of the cut-off valve.

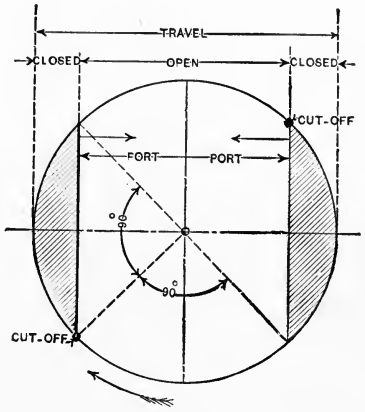


Fig. 54.

The cut-off valve sometimes slides on an anchor-plate which bears against the back of the main valve, and there are passages through the anchor-plate and the main valve which conduct the steam to each cylinder-port separately. The main objection to this arrangement is that the anchor-plate is unnecessary, for the cut-off valve may as well slide on the back of the main valve directly.

THE CUT-OFF VALVE ON BACK OF THE MAIN VALVE.

Fundamental Principles.

In order to fully understand the working of a cut-off valve on the back of the main valve certain fundamental principles should be well understood. First, it should be recognized that

the action of the cut-off valve depends entirely on its motion **relative to the main valve;** and if this relative motion can be represented abstractly in a simple manner any reference to the motion of the main valve will be useless.

In Fig. 55 B is the center of the main eccentric, and E is the center of the cut-off eccentric, and these centers are connected by an imaginary straight line. The two points B and E form a connected system of points, which moves around the center A, and, at the same time, the system revolves about itself, or about some point in the system; thus: point B describes a circle around point A, and, at the same time, point E describes a circle around point B. Four positions of points B and E are shown by full lines, and dotted lines indicate the positions E would take if it did not revolve about B; in which case the cut-off valve would evidently not slide on the main valve at all, and there would be no cut-off action. This action, therefore, depends on the rotation of E around B, and is not in the least effected by motion of B about A; and the relative motion of the cut-off valve will be as if it worked on a stationary seat, and was driven by an eccentric of eccentricity equal to distance B E. The upper circle, Fig. 55, represents the path of this equivalent ideal cut-off eccentric, and corresponding and coincident positions of the eccentrics in both figures are indicated by the notation. Note that **corresponding positions of the cut-off eccentric arm, from B as a center of rotation in both figures, are parallel.** When the equivalent eccentric is at mid-throw the cut-off valve is in its central position relative to the main-

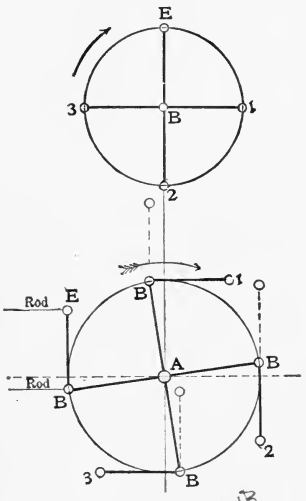


Fig. 55.

valve, and if the cut-off edges were then "line and line," the cut-off action would evidently be the quickest possible; and the fallacy of the common statement that the quickness of the cut-off is due to the fact that main- and cut-valve move in opposite directions, becomes evident, for, as in the present case, the cut-off may become quicker if the main valve does not move at all. The superiority of the independent cut-off lies in the facts that no restriction of the main port-opening is required, and that the cut-off may occur when the virtual, or equivalent, cut-off eccentric is near the middle of its throw.

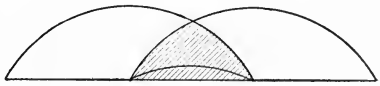


Fig. 56.

The independent cut-off action, as compared with the single-valve action, is shown approximately in Fig. 56, where the shaded area represents the port-opening for one-quarter cut-off with separate cut-off valve, and the flat arc represents the corresponding single-valve opening.

THE MEYER CUT-OFF.

Fig. 57 shows the Meyer cut-off, which has been in use since 1842. Adjustable slides work on the back of the main valve, and are driven by a separate fixed eccentric. The slides are shown close together, which is the adjustment for latest cut-off. A and B are cut-off edges, and the distance they are apart constitutes negative lap, which is called "space." By turning the screw the slides are separated and the space is diminished, which has the same effect as increasing the lap on other valves; that is, it brings about an earlier cut-off. The stem of the screw extends through the back of the steam-chest; and it

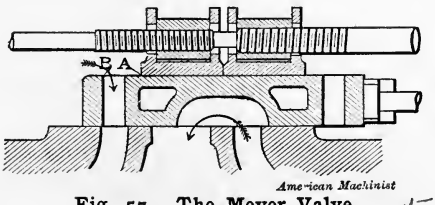


Fig. 57. The Meyer Valve.

may be turned, without stopping the engine, by means of a hand-wheel mounted on a sleeve. The main valve has an extension on each end, which contains the passage through which the steam is conveyed to the main cylinder-port; otherwise the face of this valve is constructed as that of a common slide-valve, designed to cut-off at about three-quarters of the piston-stroke.

THE MEYER CUT-OFF DIAGRAM.

The essential data required for the design of the Meyer cut-off mechanism, can readily be obtained by paying close attention to fundamental principles. The lower circle, Fig. 58, represents the path of the main eccentric. At the end of the horizontal diameter, and with radius equal to the lead, make a small

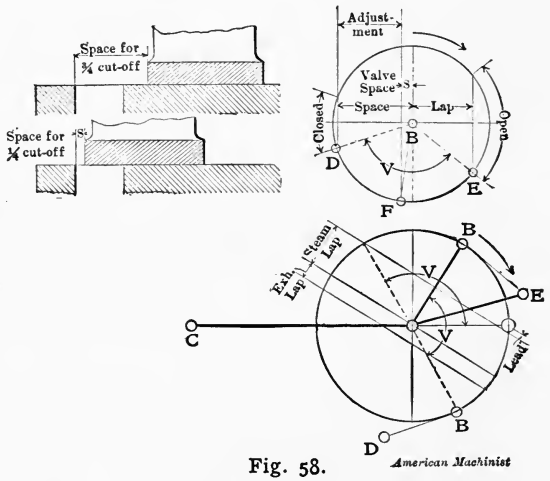


Fig. 58. *American Machinist*

circle, as shown, and draw a chord touching this circle, and whose projection on the diameter of the eccentric-circle bears the same proportion to the whole diameter as the distance traversed by the piston before steam is cut off by the main valve bears to the whole stroke, and which, in the present case,

is supposed to be as three to four. A radius at right angles to this chord indicates the position of the main eccentric, B, when the crank is at its dead center at C. With a radius somewhat smaller than that of the main eccentric, draw a circle directly above, as shown. This circle represents the path of the equivalent or ideal cut-off eccentric. The extreme range of adjustment of the cut-off valve, relative to the port on the back of the main valve, is indicated to the left, where the cut-off valve is shown in its middle position, adjusted for earliest and latest cut-off respectively.

First fix the cut-off position of the equivalent eccentric when the valve is set for latest cut-off. For reasons which will appear afterwards, this position should be near the extremity of the throw; let it be at D in the upper circle. When the piston has traversed three-quarters of its stroke, the equivalent cut-off eccentric is at D, at the instant that it closes the port; and, therefore, by marking off from D backwards the three-quarter stroke angle V, the point E is located, which marks the position of the equivalent eccentric when the crank is on its dead center, at the commencement of the stroke; and by transferring the arm BE to a parallel position on the main-eccentric-circle, as shown, the proper location of the cut-off eccentric relative to the crank-shaft is obtained. The speed of the engine is, in the present case, supposed to be regulated by a throttling governor, and the point of cut-off will only be varied occasionally, to suit the load or steam-pressure; and it is, therefore, not necessary to cut off the steam earlier than at one-quarter of the stroke. Make angle EF equal to the quarter-stroke crank-angle, and F will mark the one-quarter cut-off position of the equivalent eccentric, and the distance of F from the vertical center-line gives the negative lap or valve-space for earliest cut-off, while the valve-space required for latest cut-off is given by the horizontal distance of point D from the center line; and the horizontal distance between these

two points represents the entire range of adjustment of the cut-off valve in the present case. If earlier cut-off is required, the valve-space must be made smaller. For cut-off at one-eighth of the stroke, the cut-off edges would coincide, and for still shorter cut-off, the valve would overlap the port in its midled position. The motion is quickest at the middle of the throw, and the quickest cut-off is that which occurs at or near mid-throw of the equivalent eccentric. The radius of the eccentric-circle may represent maximum speed of relative motion at mid-throw, and at other points it is represented by vertical ordinates to the horizontal diameter. It will be seen that sharper cut-off could be had for the entire range by moving point D farther back on the circle; but this would also throw point E farther back, and would increase the diameter of the cut-off eccentric; and as it so happens that a sharp cut-off earlier in the stroke is of more consequence than later in the stroke, the arrangement here shown is generally adopted. Thus it appears that the design of the Meyer valve-gear is restricted by a desire to keep down the size of the cut-off eccentric; and it will also be readily seen that the eccentrics cannot very well be of same diameter, unless the range of cut-off be limited to five-eighths of the stroke. The latest cut-off may occur before the main port is closed, and in that case the short period during which the cut-off valve is closed, as shown on the diagram, must be taken into account; for otherwise it may happen that the cut-off port is open again before the main port is closed. The location of the cut-off eccentric, when the main port is closed, is found by marking off the corresponding crank-angle from point E forward.

It will probably be admitted that the graphical method here shown is simpler than that invented by Dr. Zeuner; but the greater advantage of the method here presented lies in the fact that the geometrical construction is less artificial, and may readily be reasoned out, and may therefore be retained in the mind more readily than any process that must be learned

mechanically, and which requires a somewhat complex geometrical construction which must be produced before the problem is considered. For the sake of simplicity and clearness, it seems better to use the simple Sweet diagram for the main valve, and treat the cut-off valve separately; for nothing is gained, and much may be lost, by drawing so many lines and circles in one figure. It should also be considered that the planning of the valve motion, and not the drawing of the diagrams, takes the designer's time. It does not pay to rub out many lines and draw others in their place; for paper is cheap, and the drawing of a fresh diagram requires but little time, and this time is of little consequence if the valve-gear to be designed is worth anything.

LIMITATIONS OF THE MEYER CUT-OFF.

The Meyer valve-gear is admirably adapted for hand adjustment, but the adjusting screw may be operated by the governor. This may be accomplished by means of a rack and pinion, but the motion required and the friction of the screw are unfavorable conditions for a satisfactory regulation of the speed of the engine; and the difficulty is increased by the fact that if the range of cut-off is to extend to the beginning of the stroke, the adjustment of the slides will be about twice that required if the cut-off is beginning at one-quarter of the stroke. By making the right-hand screw and left-hand screw of different pitch, the cut-off can be made to conform closely to the irregularity of the piston-motion caused by the angular motion of the connecting rod.*

THE RIDER CUT-OFF VALVE.

If the cut-off ports on the back of the main valve are made converging, and the cut-off edges of the valve are made to conform to this, the lap or valve-space may be varied by a lateral

* The effect of the angular motion of the connecting-rod is explained in the last chapter.

adjustment of the cut-off valve. This principle holds if the valve surface is made cylindrical, in which case the valve-seat becomes concave, and the cut-off edges become right- and left-hand screw-lines of steep pitch. The adjustment for earlier or later cut-off is then effected by a partial rotation of the cut-off valve, which is done automatically by a governor of the Porter type, by means of a rack which engages a pinion on the valve-stem. This form of valve-gear was invented by A. K. Rider of New York, and it has been applied to quite a number of engines; but at the present time the combination of a shaft governor and a single balanced valve is preferred.

CUT-OFF VARIED BY ROTATING ECCENTRIC ON THE SHAFT.

The cut-off valve may be a simple slide with inner or outer cut-off edges, driven by an eccentric loosely journaled on the engine-shaft and connected with a shaft governor, which automatically changes the cut-off in relation to the piston-stroke, by rotating the eccentric on its journal.

Fig. 59 shows a main valve similar to that in the Meyer system. On the back of this valve is a slide, which cuts off

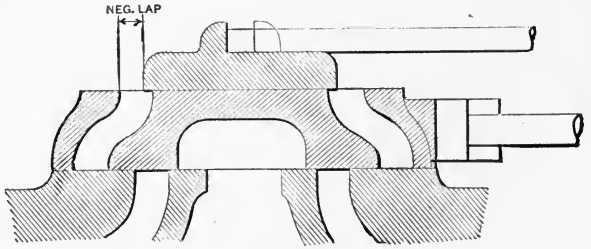


Fig. 59.

the steam by its outer edges. Both valves are shown in their central position, in order to show the lap, and the **negative lap,** or space, of the cut-off valve will be noticed. The main valve operates as a common slide valve, cutting off steam at about three-fourths of the piston-stroke.

Fig. 60 represents the relative motion of a cut-off valve. B may represent the center of the main eccentric, supposed stationary. Let the relative throw of the cut-off eccentric be represented by the smaller circle, then the point of cut-off will be at D; but if the throw is increased, as shown by the larger circle, then the point of cut-off will be at F. E and F mark different positions of the cut-off eccentric at the beginning of the piston-stroke. When this eccentric is in position E it will cut off steam at three-fourths of the stroke, and in the position F no steam will be admitted to the cylinder. For intermediate cut-offs the eccentric would be located somewhere between points E and F. Angle V measures 120 degrees; and from E to F the eccentric moves through an angle of 120 degrees minus angle U or nearly 90 degrees. It will be seen that an increase of the throw diminishes the angular adjustment. If, therefore, the cut-off eccentric be so located in relation to the main eccentric that when it is rotated on the engine-shaft the distance between the eccentric centers is increased while the cut-off eccentric is angularly advanced, a considerable range of cut-off may be obtained.

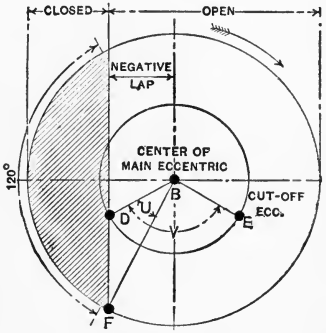


Fig. 60.

Diagram 61 shows how to find the position of the eccentrics, the relative motion of the cut-off eccentric, the lap of the valves, etc. The lower circle represents the throw of the main eccentric, which is fixed on the shaft, and the dimensions of the main valve are shown by Sweet's parallel lines, which also determine the position B of the main eccentric. The upper circles represent the motion of the cut-off eccentric around the center of the main eccentric. The smaller circle represents the relative throw of the cut-off eccentric for latest cut-off, D being the

point of cut-off. Make angle D–E 120 degrees, and draw B–E parallel B–E in the lower figure. This fixes the location of the cut-off eccentric E. Draw large upper circle to make shaded part include a little more than 120 degrees of the circle, and make B–F parallel B–F in the lower figure. This determines the position F of the cut-off eccentric at shortest cut-off, and E–F is the angle it must be rotated on the engine-shaft. Note that radius B–E must be a little more than the negative lap in order to keep the cut-off port closed a short time after the main port is closed. Angle V represents the range of cut-off, which in this case is 120 degrees of crank angle, or three-fourths of the stroke.

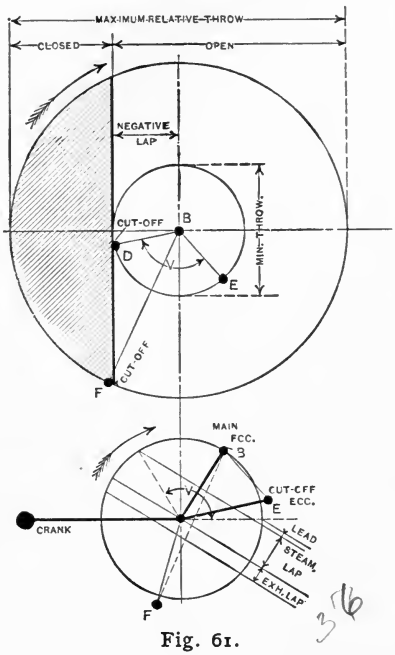


Fig. 61.

This construction gives a very sharp cut-off early in the stroke, and a large port-opening. The latest cut-off is slow, but at midstroke it is very satisfactory. The cut-off eccentric is of about the same size as the main eccentric; but its relative throw for short cut-offs becomes much greater. The cut-off is regulated by a shaft governor, and the greater rate of adjustment for short cut-offs is conducive to close regulation.

The rotative movement E–F, in the lower figure, should not be greater than here shown, for otherwise it will be very difficult to construct connections between the governor weights and the eccentric, which will be durable and sufficiently effective in the extreme positions.

The cut-off valve shown in Fig. 62 has **considerable lap,** and it cuts off by inside port edges. It is driven by an eccentric journaled on the engine-shaft, and controlled by a shaft governor, identically in the manner just described.

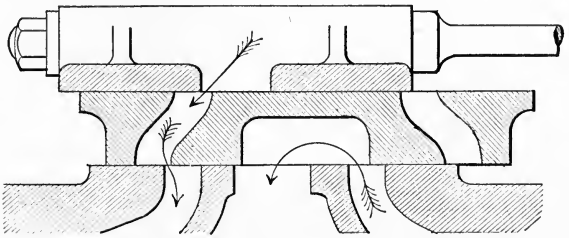


Fig. 62.

The valve diagrams are shown in Fig. 63. The lower circle represents the path of the main eccentric, which is fast on the shaft, and also the circular path of the cut-off eccentric, both having the same throw. The crank is at its dead center, at C, the main eccentric is at B, and the cut-off eccentric is at E. This is the natural position of the cut-off eccentric relative to the main eccentric when the engine is not running, or before the governor begins to act, and when steam is cut off a little before three-fourths of the stroke. When the proper speed is attained the governor becomes active, and turns the eccentric forward. When the cut-off eccentric is advanced to the position F steam is cut off right at the beginning of the stroke, and intermediate positions of the cut-off eccentric cover the whole range of cut-off. The eccentricity of the cut-off eccentric in regard to the shaft is fixed, but in regard to the main eccentric it varies between the two extremes represented by distances B–E and B–F. The upper circles represent the variable throw of the cut-off eccentric in relation to the center of the main eccentric, B; the cut-off action being as if the valve-seat were stationary and the throw of the eccentric as represented by the upper circles. F marks the position for earliest cut-off, the

crank being on its dead center, as shown in the lower figure. Make B–F parallel B-F, and through F draw a vertical; this gives the lap of the cut-off valve. With radius B–E strike a circle, and its intersection with the lap-line, at D, marks the

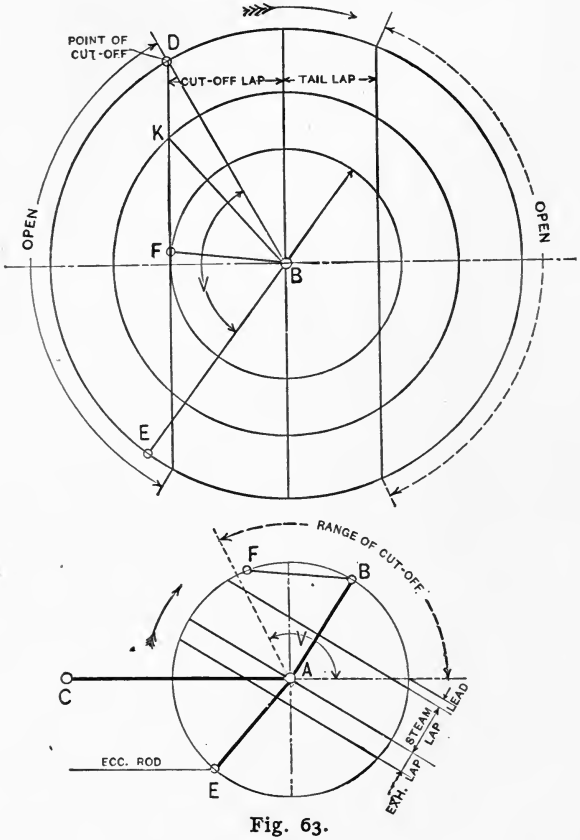


Fig. 63.

latest cut-off relative to the center line of action, which is supposed to be horizontal. Draw B-E parallel B–E, and angle V is equal to the crank-angle at latest cut-off, that is, it represents the extreme range of cut-off, which in this case is a little less than three-fourths of the stroke.

Point E marks the position of the cut-off eccentric at the commencement of the stroke, and it should fall outside the lap-line, as shown; for the port should be open before the commencement of the stroke. The moment the eccentric passes the lap-line steam is cut off, and the rapidity of valve-motion at this point is represented by the distance above the horizontal center line. At the beginning of the piston-stroke the cut-off action is slow, but it increases rapidly. K marks the cut-off at one-fourth of the piston-stroke, and at this point the motion is as quick as that of the Meyer cut-off, and the port-opening is more than four times that obtainable with a single valve. For very early cut-off the port-opening becomes exceedingly small, and for this reason a valve with negative lap is probably preferable. The small port-opening "wiredraws" the steam, reducing the initial steam-pressure in the cylinder, and therefore, at very early cut-off this valve would not admit enough steam to run the engine alone, in other words, there would be no early cut-off. It is as yet unsettled whether this would be detrimental or beneficial to steam economy. At any rate, it will only be of consequence when the engine is underloaded.

When laying out the valve-motion the following must be observed: The cut-off valve should preferably open the port a little before commencement of the stroke; that is, point F should be a little above point B. The movement around the shaft should not exceed a certain limit, for otherwise it will be difficult to construct proper governor connections. The range of cut-off should not be much less than three-fourths of the stroke. It will be noticed in Fig. 62 that the tail-end of the cut-off valve opens the port a second time, which in no case must happen before the main port is closed. To guard against this a vertical line is drawn in the upper diagram, to the right from the center, at a distance equal to the tail-lap of the valve, and the intersection of this line with the eccentric-circles shows the angle turned by the crank before the port is uncovered.

The great friction of the common D-valve makes it unsuited for single-valve automatic cut-off engines, but in combination with a cut-off valve it operates under different conditions. The pressure in the exhaust-cavity is but little more than that of the atmosphere, while the corresponding area on the back of the valve is exposed to full steam-pressure. It is, therefore, desirable to reduce the exhaust-area as much as possible, and to shorten the travel. In single-valve automatic engines, and in locomotives, the small port-opening by short cut-off must be taken into consideration, but with a fixed cut-off at three-fourths of the stroke the conditions are quite different. The travel can in that case be very short; for the port opens quickly at the beginning of the stroke, and it is not necessary that it uncover the port entirely. The shorter travel makes it feasible to reduce the exhaust-cavity, and when properly proportioned the common D-valve with a riding cut-off will possess some points of merit not claimed for the balanced valve.

GRIDIRON VALVES WITH INDEPENDENT CUT-OFF.

Flat gridiron valves are used with cut-off valves of the same description. A longitudinal section through a pair of such valves is shown in Fig. 64. The cut-off valve is supposed to

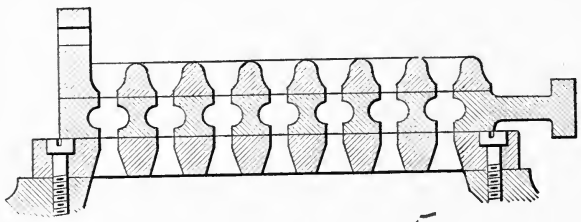


Fig. 64. 3 5

be in its central position in relation to the main valve, and the space between the bars represents negative lap. These valves, cut-off by one port-edge only, which may be either one of the two edges. They are used in four-valve engines, and there is

one set at each end of the cylinder, and there can only be one cut-off for each rotation of the eccentric; but the opposite edge of the adjoining bar will cover the port part of the time while the main port is closed. This is merely incidental to this particular valve construction; for, it being desirable to have a great number of ports, the bridges are made just wide enough to avoid uncovering the port a second time in one stroke of the valve, that is, moving in one direction. The port will therefore not remain open as long as indicated by diagram 61, but there will be a period of closed port on the right hand of the center of the upper circles corresponding to the period of closed port on the left side. The second closing will in no case take place before the main valve has closed the cylinder-port; and it is rather an advantage to have the cut-off port closed during part of the exhaust period, for it may prevent leakage into the exhaust.

If the cut-off must be carried as far as three-fourths of the stroke, and there is a rectilinear transmission from the eccentric to the valve, then the width of the bridges should be about two and three-quarter times the width of the port. There are three points which keep the design of these valves within narrow limits, viz., the size of the eccentric, the angular adjustment of the eccentric, and the parallelism of the eccentric-rods. If the cut-off is not to be later than at five-eighths of the stroke more satisfactory proportions and shorter travel may be had.

BEGTRUP'S ECCENTRIC.

If the cut-off eccentric were mounted on a journal concentric with the center of the main eccentric its relative throw would be constant and short, but there are great practical difficulties in the way of this scheme when applied to large engines. It would make a very large eccentric on a very large journal, and the friction and inertia would be too great for a sensitive governor. The bearing, movement and throw

of the cut-off eccentric may be reduced in a practical manner now to be described.

In Fig. 65 B is the center of the main eccentric, and E and F mark the positions of the cut-off eccentric for latest and earliest cut-off respectively. It will be noticed that the throw is not greatly increased in the early cut-off position if the eccentric is movable on the curved track E–F. It could be made so by swinging it from a point P; but this point would be inside the engine-shaft, and cannot be used as a fulcrum. Identically the same motion may, however, be given the eccentric by suspending it from two short links, one on each side of the shaft, as shown in Fig. 66. To bring the eccentric central over the shaft, as here shown, it will be necessary to swing it around the center B through an angle of 35 degrees, and this will not alter the valve-motion if the eccentric-rod is also turned 35 degrees from the horizontal, or if it makes that angle with the main eccentric-rod. To this there would be no serious objection; for the rod can be made short, and an angular rocker would transmit the motion straight to the valve. In its central position this eccentric becomes concentric with the shaft, leaving the valve stationary. The cut-off is then effected by the motion of the main valve only. The relative throw is considerably less than when the eccentric is moved around the shaft, and this short throw makes it suitable for multiported valves. It is not necessary that the cut-off eccentric be swung centrally over the shaft. If it is made the same size as the main eccentric the opening for the shaft may be to one side from the center, suffi-

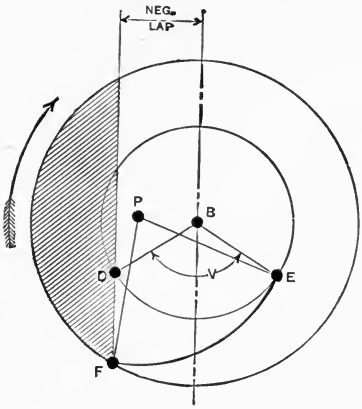


Fig. 65.

ciently so to make the eccentric rods parallel, and this does not make the throw in relation to the main eccentric any greater. A governor of the simplest description may be used with this eccentric.

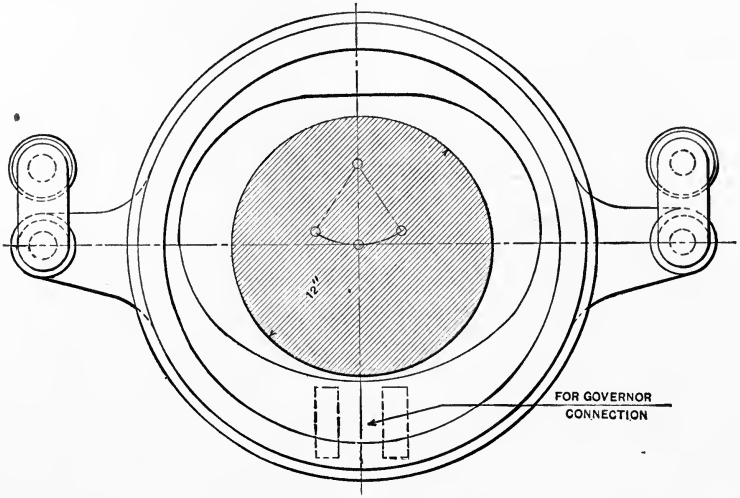


Fig. 66. Begtrup's Eccentric.

CUT–OFF WITH CONSTANT TRAVEL ON MAIN VALVE.

The cut-off eccentric may be journaled concentric with the center of the main eccentric, and then the travel of the cut-off valve in relation to the main valve will be constant ; but this arrangement requires an enlargement of the cut-off eccentric, unless the cut-off can be worked from the end of the engine-shaft.

Fig. 67 shows how a somewhat radical improvement becomes possible by the interposition of a bell-crank lever. It is here shown as applied to a vertical engine, but it is also applicable to horizontal engines. The crank is on its upper dead center, at C, and the main eccentric is at point B, which is also the center of the journal of the cut-off eccentric. The center of the cut-off eccentric is at E, and it is capable of rotative adjustment

around B to point F. When at this point, steam is cut off right at the commencement of the stroke, when the crank is on its upper dead center, as shown; and the latest cut-off, at three-quarters of the stroke, is obtained when the eccentric is swung back to position E. The valves are shown in Fig. 68; they are

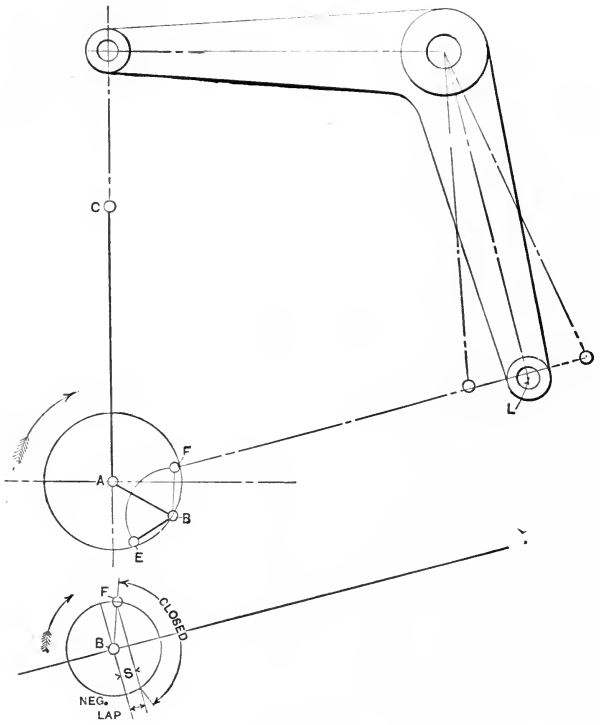


Fig. 67.

placed centrally in order to show their lap, but when connected with the valve-gear they are never in their central positions at the same time. The cut-off valve is actuated by a bell-crank lever which effects a reversal of the eccentric-motion, and so steam is cut off by the outer edges of the valve. The main valve is a flat plate with a rectangular exhaust opening in the

center and cut-off ports near the ends. The cut-off valve has a pressure balancing-ring on its back, which relieves the main valve of the greater part of the steam-pressure; and as the compression is constant a very satisfactory degree of balancing may be obtained without causing the valve to leave its seat under any circumstance, except when forced out by a body of water in the cylinder. The travel of the main valve is constant, and the travel of the cut-off valve, **in relation to the main valve,** is also constant. The travel of the cut-off valve in relation to the steam-chest cover is not constant, but it will appear from an inspection of Fig. 67 that the maximum throw of the cut-off eccentric is less than the throw of the main eccentric. Near the middle of its rotative adjustment its throw is greatly reduced, and the valve could be made to stand still in its middle position if the radius of the curved path were made a little longer. It is noticeable that the shortest valve travel occurs when it is cutting off steam at one-fourth or three-eighths of the stroke, under which condition the engine is supposed to do the greater part of its work. Slide-valve designs involve, to a great extent, the reconciliation of conflicting requirements, and absoute perfection seems unattainable or incompatible with great simplicity; but the combination here shown has many good points, and it would often be serviceable.

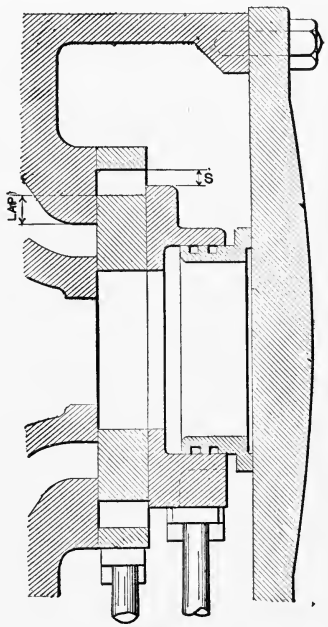


Fig. 68.

To get the lap, etc., of the cut-off valve, draw a circle with radius BF, as shown, and make BF parallel BF. Draw a center

PALLIDIE LIBRARY OF THE UNIVERSITY OF

line parallel FL, and at right angles to this two parallel lines through B and F in the lower circle: the distance between these is negative lap or valve-space, which added to the radius gives full port-opening, provided both arms of the bell-crank lever are of same length. The diameter of the smaller circle represents the relative motion of the valves, which is constant; and the smaller part of the circle, cut off by the lap-line through F, represents the period during which the cut-off port is closed, and which should be a little in excess of the period during which the main port is open; it should in this case encompass more than a three-quarter stroke crank-angle. As the throw of the equivalent cut-off eccentric is constant, and its port-opening is greater than its half-throw, it follows that a comparatively short relative motion is required, to give as much opening as the main valve which has considerable positive lap. An unusually large eccentric is therefore not needed, but it must need be somewhat larger than the main eccentric, unless the diameter of the shaft can be reduced. The main eccentric-rod is supposed to be vertical, and it is necessary that the eccentric-rods form an angle between them as shown, in order to get the desired valve-motion.

THE BUCKEYE VALVE-GEAR.

Mr. J. W. Thompson has designed a simple and ingenious valve-gear, which connects the motion of the cut-off valve with that of the main valve in such a manner that the travel of the cut-off valve, in relation to the main valve on which it slides, becomes equal to, or proportional to, the throw of the cut-off eccentric, irrespective of its position relative to the main eccentric. The cut-off eccentric may, therefore, be journaled directly on the engine-shaft; and the point of cut-off may be changed by a rotative adjustment of the cut-off eccentric (regardless of the position of the main eccentric), and as the throw of the cut-off eccentric is constant **the relative motion of the cut-off valve will also be constant.** There are two rocker-arms, one for the main

eccentric and one for the cut-off eccentric. The cut-off rocker is carried by the main rocker, and it is fulcrumed mid-way between the valve connection and the fulcrum of the main roeker. It is a counter-arm rocker of same total length as the main rocker ; and the lower end, to which the eccentric-rod is pivoted, is in line with the fulcrum of the main rocker : and, this being stationary, it follows that any motion transmitted by the cut-off eccentric-rod to the lower end of the cut-off rocker appears reversely relative to the upper end of the main rocker, irrespective of the position of the main rocker ; that is, the movement of the upper end of the cut-off rocker **relative to the main rocker** is equal to the throw of the cut-off eccentric ; and as both valves receive their motion from the upper ends of the rockers, it follows that the travel of the cut-off valve, relative to the main valve on which it slides, is constant ; and that steam is cut off earlier or later according as the cut off eccentric is advanced or moved back on the shaft.

The valve used on the Buckeye engine, in connection with the compound rocker just described, is shown in Fig. 69. The

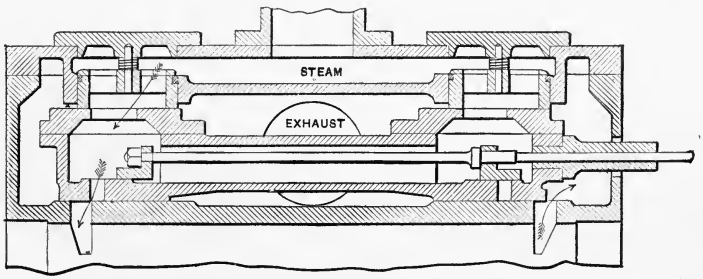


Fig. 69. The Buckeye Flat Valve.

body of the main valve is hollow, and two narrow cut-off valves slide on inside valve-faces, next to the cylinder valve-faces. The steam enters a hollow part of the steam-chest cover, whence it passes through sliding ring-joints to the inside of the hollow

valve, and from there through the cut-off ports, over the inside edges of the main ports, to the cylinder. The exhaust escapes over the outside port-edges to the space surrounding the main valve and which connects with the exhaust-pipe. The main-valve stem is hollow and the cut-off stem passes through it. The proportions here shown are approximately correct for a long-stroke engine, but minor details are not shown. This construction has the advantage of short and direct steam-passages and small cylinder-clearance; and as the unbalanced pressure is quite moderate, the wear of the valve-faces cannot be of much consequence, and at the same time this pressure, in conjunction with a constant travel, insures permanent tightness — unless the faces are cut by solid matter carried over by the steam. As the hot steam inside the main valve is surrounded by exhaust-steam, some heat may be lost by reëvaporation of moisture contained in the exhaust-steam; but it is noteworthy that the builders have not found it necessary to make any change in this respect since the introduction of the valve, some twenty-five years ago.

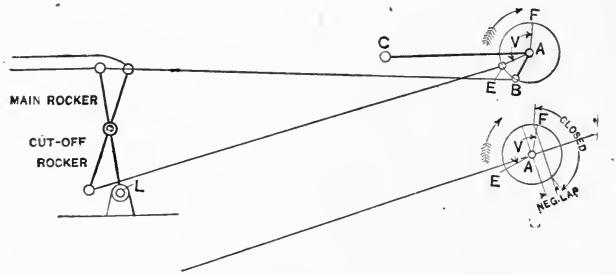


Fig. 70.

Fig. 70 shows the principles of construction of the valve-gear. The center of the engine-shaft is at A, and the crank is on its dead center, at C. The main eccentric is at B, and the cut-off eccentric is at E, and the compound-rocker is represented by two crossed lines. The lap of the main valve, its point of cut-off, etc., can easily be determined by Sweet's valve-

diagram, remembering that the location of the eccentric must be diametrically opposite that obtained from the diagram, on account of the inward position of the steam-edges. The path of the cut-off eccentric is represented by a separate circle. Through the center of this circle draw a line parallel to AL; and at right angles to this, in a distance equal to half the width of the cut-off port, draw a chord, as shown. The distance from the center represents negative lap, and point F marks the position of the cut-off eccentric for earliest cut-off. The smaller arc, cut off by the chord, represents the period during which the cut-off port is closed; and it is only necessary to observe that this port must not open before the main port is closed. If, for instance, the main port is closed at three-quarters of the piston-stroke, the cut-off valve may close the cut-off port a little before the commencement of the stroke, and keep it closed till after three-quarters of the stroke is traversed; and no steam will in that case be allowed to enter the cylinder.

If great accuracy is required, the varying angularity of the eccentric-rod may be taken into account; and the negative lap or valve-space is determined accurately by drawing the lap-line through F at right angles to FL above, which increases the valve-space slightly; but the first construction is satisfactory, for its error is small and on the safe side.

PISTON–VALVES WITH INDEPENDENT CUT–OFF.

Piston-valves of the larger sizes must fit loosely in the valve-cylinder, for otherwise they will occasionally become very tight. They are, therefore, provided with elastic packing-rings similar to those used on the main piston, and these rings will cause some wear, and if the travel is variable, uneven wear may be expected; and for this reason it is claimed that piston-valves with constant travel wear better than those that regulate the steam admission by varying their travel; and as a relatively

constant travel is obtainable with the independent cut-off gear, such gear may profitably be used in connection with piston-valves; and as these valves are in perfect equilibrium under any steam-pressure, they may have a long travel without excessive wear and without any detrimental effect on the governor; and the increased travel may obviate the necessity of multiple ports, which in any event is of questionable utility; for it increases the chances for leakage.

Fig. 71 shows part of a piston-valve combination, adopted by the Buckeye Engine Company. The cut-off valve is inside the main valve, and the arrangement of steam and cut-off ports is the same as that of the flat-faced valves, Fig. 69; but the travel is longer, and the exhaust steam is discharged at the

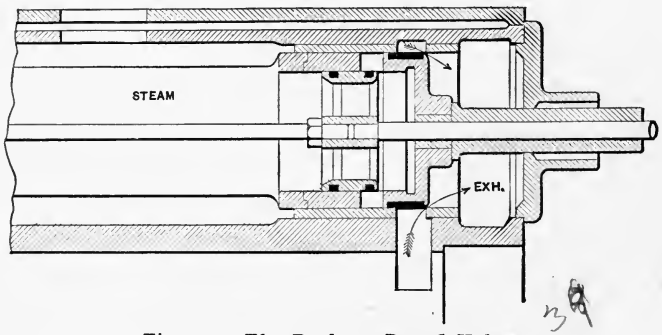


Fig. 71. The Buckeye Round Valve.

ends, and does not come in contact with the hot surface of the main valve, except at the ends. The packing-rings are in the best possible positions to prevent leakage; and both the main and cut-off ports are bridged in a number of places to afford ample bearing for the rings when they pass over the ports.

CUT-OFF VALVE WORKED BY MEANS OF A LINK.

The cut-off valve may be worked by a rocking-link as indicated in Fig. 72. In that case the cut-off eccentric E is fixed

on the shaft, and the travel of the cut-off valve is varied by moving a block up and down in the slotted link ; which may be done automatically by connecting it with a governor of the

Fig. 72.

vertical type. By shifting the link-block the throw is increased or diminished, precisely as if the eccentric were shifted on a radial line from the center of the shaft. In Fig. 73, lower circle, the main eccentric is at B, and the cut-off eccentric is at E, and the crank is at its dead center, at C. When the link-block is in its lowest position, the valve-motion is as if the eccentric were at F and its horizontal motion were communicated straight to the valve ; and shifting the link-block to its upper position corresponds to a radial displacement of the eccentric to point E′ ; and in this position BE′ is the radius of the equivalent cut-off eccentric, as if actuating the valve on a stationary valve-seat. With radius BF strike a circle, as shown above, and make BF parallel BF. Through F draw a line at right angles to AL, Fig. 72 ; its

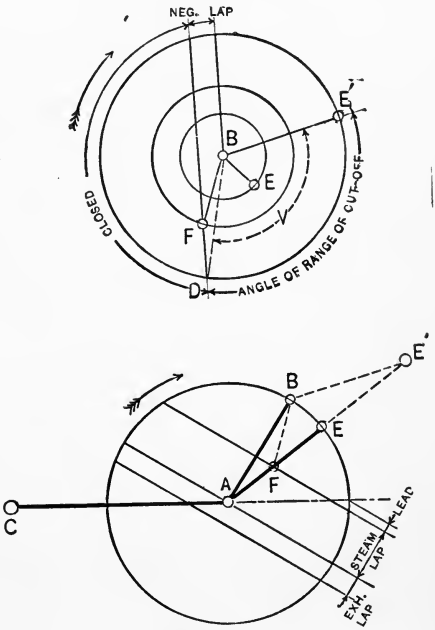


Fig. 73.

distance from the center determines the negative lap required. With radius BE′ draw a circle and make BE′ parallel BE′; the angle V determines the range of cut-off. As the ideal eccentric is supposed to move radially, its throw in regard to its virtual center of revolution at B must vary considerably; and it is observable that it is least near the middle of its radial path when steam is cut off at about one-quarter of the piston-stroke; but considering that the valve has negative lap, and cuts off near its mid-stroke, the valve-action would be satisfactory at moderate piston-speed. The cut-off valve may be a plain flat slide, cutting off steam by its outer edges; and it can be constructed with two cut-off edges at each end to operate on double-ports, though a constant relative travel is much better for double-ported valves.

As here shown the maximum cut-off will be at 3-4ths of the stroke. Much more satisfactory valve proportions can be had if the cut-off is limited to 5-8ths of the stroke.

INDEPENDENT CUT-OFF ON FOUR-VALVE ENGINES.

The McIntosh & Seymour Valves.

Fig. 74 represents a sectional view of the McIntosh & Seymour valves and valve-gear. There are two main valves at each end of the cylinder, one for admission and one for exhaust. It is plain flat gridiron valves, on detachable seats, and they move transversely to the cylinder.

In the valve-gear, links, which are used to transmit motion to the valves, are actuated by swinging rockers in such a way as to distort the motion as received from the eccentric, hastening the movement of the valve when near one end of its stroke, and at the other end causing a pause in its motion, so that while a a rapid opening and closing of the port is secured, the valve remains practically still while closed. This feature and the large number of ports in the valve reduce the stroke necessary to give

full port opening to from one-half inch up to one and one-half inch for cylinders of the largest size; and, since the movement of the valve takes place chiefly when it is open and relieved of the pressure of the steam, the wear and also the power required to

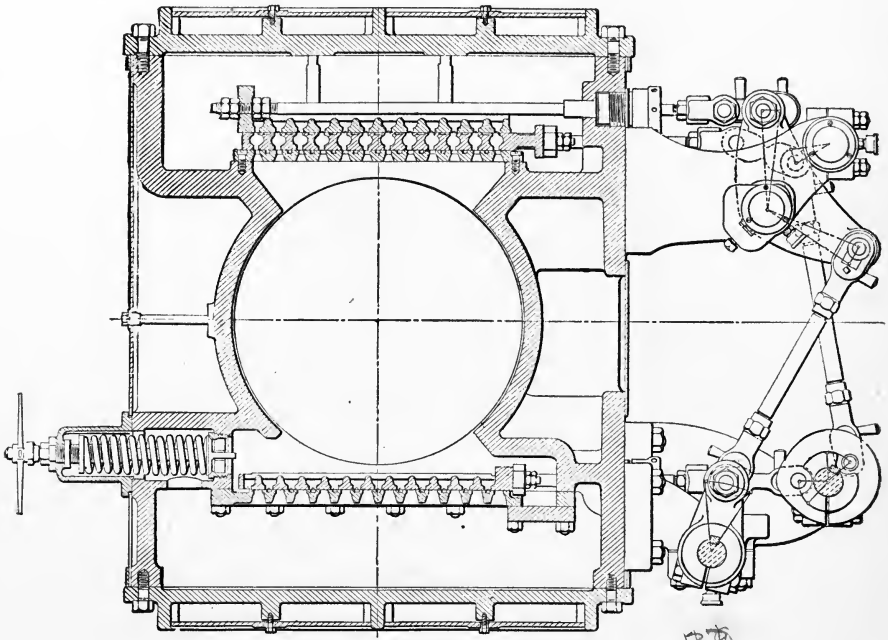


Fig. 74. McIntosh & Seymour.

operate the valve is reduced to a small amount. And since the bars or bridges between ports are of same width on valve and valve-seat, the wear should be absolutely even on the entire surface.

On the top of the main steam-valve is a cut-off valve, operated by a movable eccentric, controlled by a shaft governor. The eccentric is journaled on the engine-shaft, and the cut-off is varied by varying the angular position of the eccentric in relation to the crank. The main valves are driven by a fixed eccentric, controlling the admission of the steam and the opening

and closing of the exhaust. The valve-gear is simply an arrangement of links, rock shafts, and slides for transmitting the motion of the eccentrics to the valves. On multi-cylinder engines the governor usually controls the cut-off on all the cylinders.

Valves of the Russell Engine.

Fig. 75 shows the valves of the Russell four-valve engine. There are two admission-valves, one at each end of the steam-chest, and on the back of these are multiported cut-off valves. There are two exhaust-valves of the semi-rotary kind placed underneath the cylinder, one at each end. Steam- and exhaust-

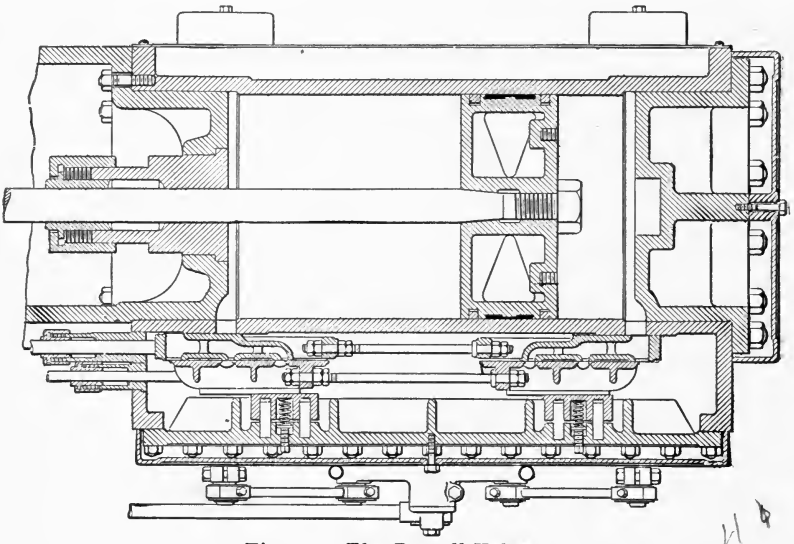


Fig. 75. The Russell Valves.

valves are driven by one eccentric, which is fixed on the shaft; and the cut-off valves are driven by an eccentric journaled on the shaft and connected with a shaft governor, which regulates the cut-off by turning the eccentric forward or backward on the shaft. Fig. 76 shows one of the exhaust-valves. It is triple ported and of small diameter. As the valve-face is at the bot-

tom, it will not become leaky when worn. These valves are actuated by means of a wrist-plate of the Corliss type, which, in conjunction with the multiplication of ports, makes large diameters unnecessary, and the valve cavity, or clearance, is thereby greatly reduced.

The plates and springs on the back of the cut-off valves, Fig. 75, are there to prevent the valves from leaving their seats when the steam-pressure in the steam-chest is reduced below that of the compressed steam in the cylinder. The compression of the springs may be increased by means of screws from the outside of the chest-cover. In this connection it should be noted that the point of exhaust closure is fixed, and that the compression corresponds to that which can be had at latest cut-off in a single-valve engine. It will, therefore, never become excessive, but it answers its purpose when the cylinder clearance is sufficiently small. The valves will only slam when the throttle valve is closed, or when used on the low-pressure cylinder of a non-condensing compound engine.

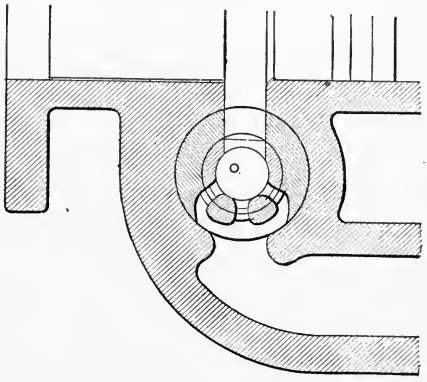


Fig. 76. The Russell Exhaust Valve.

The Buckeye Vibrating Cut-off.

Figs. 77 and 78 represent sectional front and end elevations of a valve-gear for large low-pressure cylinders, lately brought out by the Buckeye Engine Company. There are four main valves, two at each end of the cylinder, one for exhaust and one for steam. These valves are of the Gridiron pattern, and they move in a direction transversely to the cylinder. The steam-valve has a hollow cylindrical shell cast on top of it, and bored

out for a pair of cut-off valves. These valves extend the whole length of the main valve, and they receive rotative reciprocal motion from a central valve-shaft. Steam is admitted through openings in the top of the cylindrical shell, and passes through

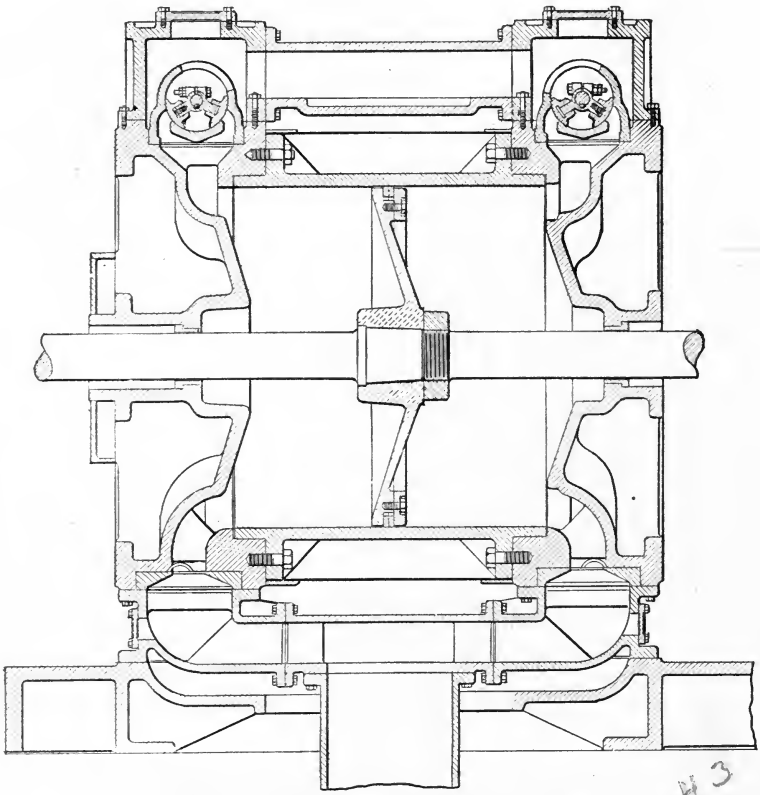


Fig. 77. The Buckeye Vibrating Cut-off.

the long cut-off ports to the ports of the main valve. The motion of the cut-off valves is similar to that of the Corliss exhaust-valves. These valves are moved by an eccentric, journaled on the engine-shaft and connected with a shaft governor, which regulates the cut-off, in relation to the piston-stroke, by rotating

the eccentric on the shaft. The stroke of the valves is constant, and earlier or later cut-off is effected by turning the eccentric ahead or back on the shaft. The main valve has a short longitudinal motion, derived from a fixed eccentric. It determines the point of admission and gives a constant lead.

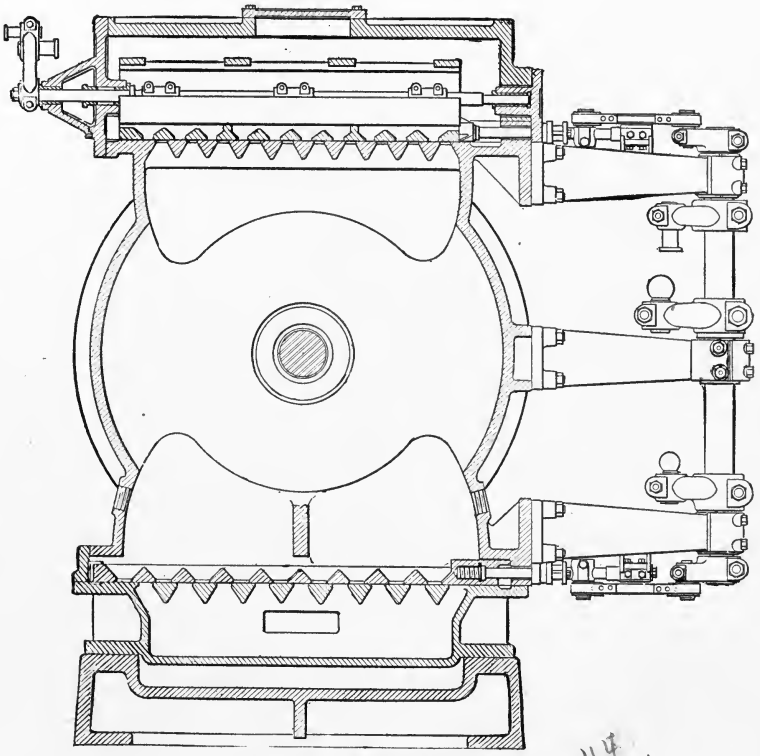


Fig. 78. Buckeye Valves.

The exhaust-valve has an invariable motion similar to that of the steam-valve, and both valves are driven conjunctively by means of a vertical rocker shaft with short arms, which, when placed in their proper relative positions, will produce the toggle motion, by which the valve motion is retarded when the ports are closed and accelerated during the opening period.

The cuts representing this valve-gear were reduced from working drawings; and the levers and valves are, for obvious reasons, not shown in the relative positions they occupy when in operation.

The cylinder here shown is the low-pressure cylinder of a tandem compound engine. The high-pressure cylinder has a valve of the Buckeye standard round-valve construction, Fig. 71. The riding cut-off in this construction, as well as the vibrating cut-off inside the gridiron valves, are varied by angular advance of the cut-off eccentric, enabling both systems to be coupled to the same eccentric and governor.

CHAPTER V.

THE SLIDE-VALVE ON PUMPS.

THE common D-valve is used on the steam-cylinder of direct-acting pumps. In these pumps steam and water pistons, or plungers, are on a common piston-rod, and the steam and water ends are connected by a frame or bars. There are two distinct types of direct-acting pumps, Single Pumps and Duplex Pumps. They are all double-acting; that is, steam is admitted to each side of the piston alternatingly. The Single style has one steam-cylinder and one water-cylinder. The Duplex consists of a pair of single pumps of same size, placed side by side, and the peculiarity of this type is that the motion of each steam-valve is derived from the piston of the other pump. By this arrangement the valve-moving mechanism becomes simple and its action positive, and these are the main reasons why the duplex type has become so popular. The more uniform motion of the water is also in its favor.

The slide-valve moves across the steam-ports, alternatingly covering and uncovering each of them; and there is, necessarily, a central position where both ports are covered (or partly uncovered). If the valve stays in that position for a moment, the pump will stop; and if the valve stops near this position, leaving only a small port-opening, the speed will be more or less retarded. Now this is what would happen if the valve-motion were derived **directly** from the piston of the same pump. The valve would slowly cover the steam-port, and finally bring the pump to a dead stop, before the end of its stroke.

The means adopted to overcome this difficulty in a single pump consists chiefly in an auxiliary piston, located in the

steam-chest and called the chest-piston. This piston engages projections on back of the main valve, so that valve and piston move in unison. Before the end of the stroke of the main piston, steam is admitted to one end of the chest-piston and exhausted from the other end; and this piston is consequently forced over to the exhaust-end, carrying the slide-valve with it.

The steam-passages leading to the chest-piston are very small; when they are uncovered the chest-piston begins to move, and after that the valve-motion is independent of the motion of the main piston, and the valve, impelled by the chest-piston, will now entirely uncover the steam-port for the return stroke of the pump.

It will be noticed that the throttling effect of the small auxiliary steam-passages is of importance, for the short period intervening between the uncovering of the small auxiliary ports and the starting of the valve is essential to a successful operation of this mechanism. The main objection is, that if any of the small passages are stopped up the chest-piston becomes inoperative.

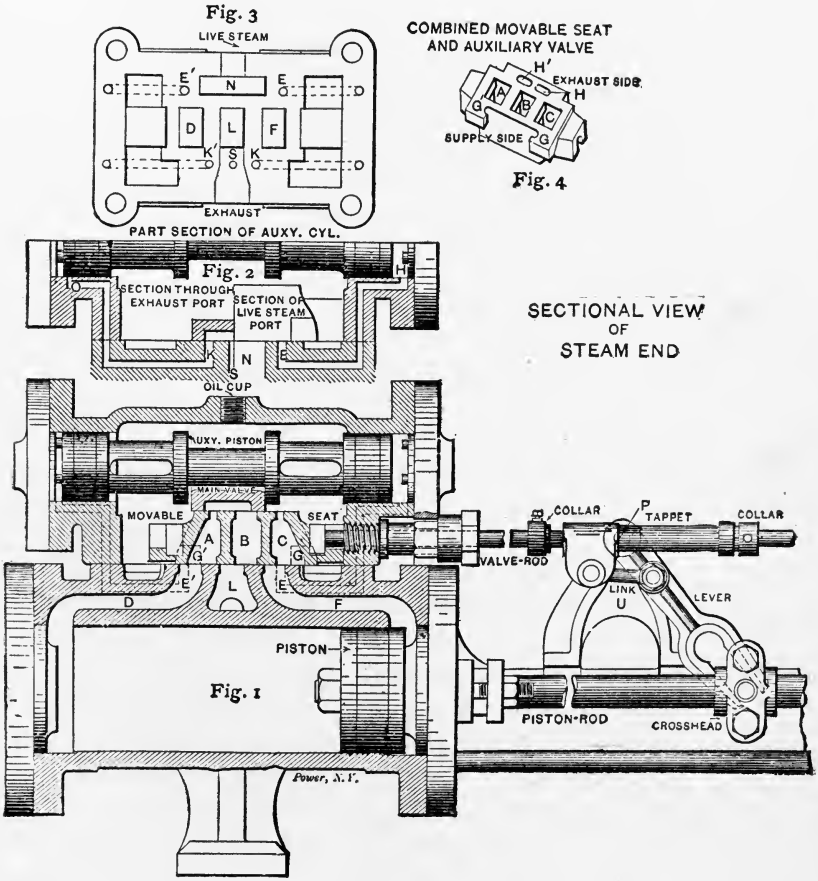
The inventor of the direct-acting steam-pump, Henry R. Worthington, found that the momentum of the moving parts of the engine was often insufficient for the purpose of throwing the valve clear across the ports, and to accomplish this he devised a combination of a steam-valve and spring. The spring being compressed by the action of the engine while the valve remained at rest, until at the proper time, by the further action of the engine, the spring was released and acted upon the valve independently of the momentum of the engine. This device was patented in 1841. Several years later he used the auxiliary piston.

The piston speed of pumps is much slower than that of a slow running steam-engine, and the steam-ports are made correspondingly small. No lap is required on the valve, and this helps to reduce its travel and size. The steam-pressure ordi-

narily sustained by these valves is, therefore, of no consequence, and the common D-form is eminently suitable for pumps.

THE BLAKE SINGLE PUMP.

The valve-gear of the Blake Single Pump is illustrated in Figs. 1 to 4 below. This is the arrangement of the boiler-feed and pressure pumps. The main valve, which controls the admission and exhaust of steam from the main cylinder, is car-



ried by the chest-piston, and moves on the back of a movable seat. This movable seat is shown, bottom up, in Fig. 4. The passages A, B, C serve as steam-ports to the main cylinder. The lugs G, G′ control the admission of steam to the chest-cylinder, and the holes H, H′ control the exhaust from that cylinder.

With the valve in the position shown, the course of the steam is through live-steam-passage N, through the port C to the right-hand end of the main cylinder, thus forcing the piston over to the left. Now, when the piston nearly reaches the left end of the cylinder the movable seat, by means described later, is shifted over to the left, so that the lug G covers the port E, while the lug G′ moves off from the port E′, thus admitting steam behind the chest-piston at the left-hand side. At the same time the hole H in the movable seat places K and S in communication, thus exhausting the steam from the right side of the chest-piston. This piston is then forced over to the right, and the port A is uncovered to live steam. When the piston is near the right-hand end of its stroke all the operations are repeated in the opposite direction.

There is one steam and one exhaust hole at each end of the chest-cylinder ; and the movable valve seat is, in effect, an auxiliary valve, which controls the admission and exhaust through said holes. The chest-piston passes over the exhaust-hole near the end of its stroke, and is thus cushioned on the imprisoned steam. The valve has no lap, a short travel, and opens simultaneously for steam and exhaust. It remains open till the piston stroke is nearly completed, when it is suddenly moved to the other extremity of its travel.

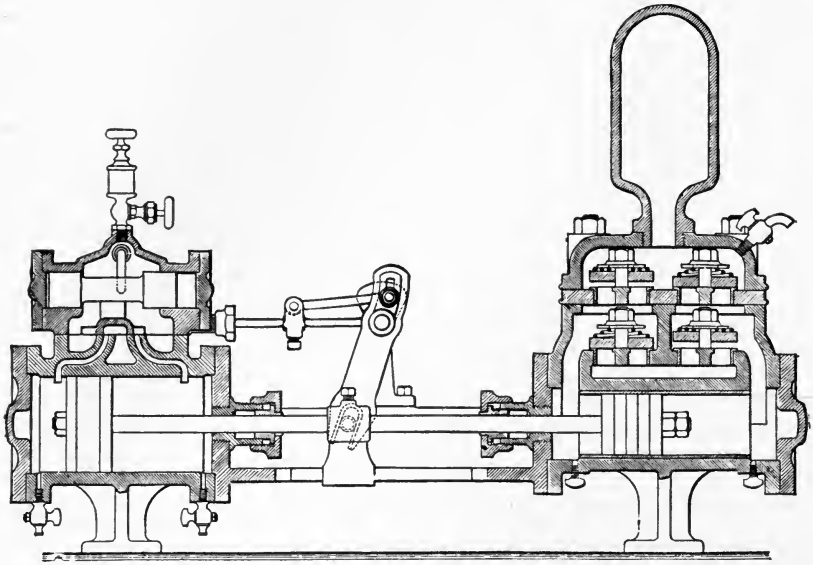
It will be seen that if means are provided to shift the movable seat, the rest of the operation is automatic. To accomplish this the piston-rod is provided with a cross-head, on which is a pin, which engages a lever fulcrumed at P. A short link connects this lever with a tappet on a sleeve, which slides freely

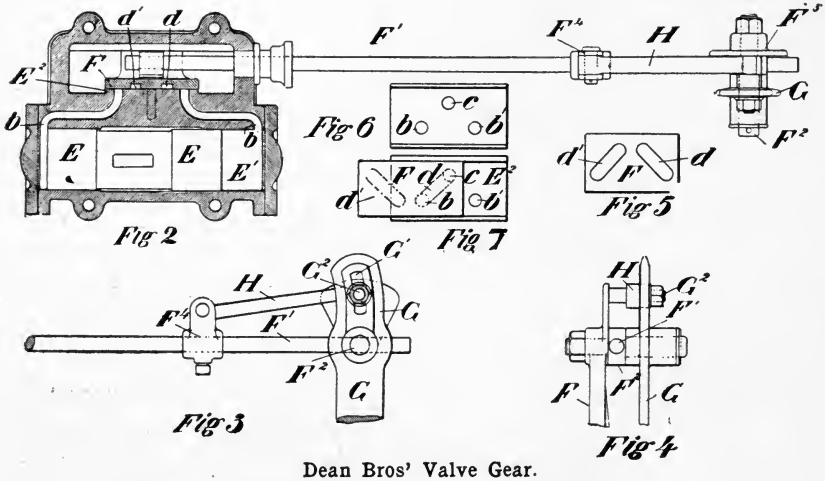
on the valve-stem. Near the end of the stroke the tappet strikes one of the two collars on the stem and moves the valve-seat. By changing the position of these collars any desired amount of lead may be given to the valve. They should be set so as to admit steam in time to stop the piston before it strikes the cylinder head. If it is stopped too soon the steam required to fill the clearance space is wasted.

DEAN BROTHERS.

Dean Bros' Single-Pump is shown on page 120. The small slide, shown in enlarged horizontal section in Fig. 6, controls the admission and exhaust of steam to and from the chest-piston. Sunk in the face of this slide are diagonal exhaust cavities d and d′. In the extreme positions of the slide these cavities connect one of the ports b, b′ with the exhaust port c, while the other port is left uncovered, as shown. The steam is admitted to one end of the chest-piston, and exhausted from the other end, forcing the piston over to one end of its cylinder; and when the slide is moved to the other extremity of its travel this operation is reversed. The chest-piston carries the main slide-valve by means of a lug on its back; and thus the main steam-ports are alternately opened to steam and exhaust, and with the stroke of the auxiliary slide properly regulated, steam will be admitted to the main cylinder in time to stop the piston at the end of each stroke.

As will appear from the illustration, the auxiliary slide receives its motion from a rocker, fulcrumed in the latitude of the valve-stem. It is a continuous motion; and the ports leading to the chest-piston are therefore closed, except at the moment the main piston is being reversed. Hence there can be no "blow through," or waste of steam, in case the chest-piston becomes worn. The stroke of the pump is regulated by lengthening or shortening the travel of the auxiliary slide, which is effected by shifting the stud in the slot in the upper end of the rocker.





Dean Bros' Valve Gear.

KNOWLES.

In the Knowles Pump there is no auxiliary slide-valve; but when the main piston has almost completed its stroke the chest-piston is slightly rotated, whereby small ports in the underside of this piston are put in communication with corresponding ports in the chest-cylinder, and thus steam is admitted to one end of this cylinder, and exhausted from the other end through small passages in the piston. The rest of the valve-motion is practically the same as that of the Blake pump, above described. The main piston-rod carries an arm which strikes a collar on the chest-piston rod, and moves the chest-piston if it should fail to start at the proper moment.

THE DAVIDSON PUMP GEAR.

The valve-gear consists of a valve, valve-pistons, valve-pin, and cam. The valve is actuated by a positive mechanical connection with the main piston-rod, and by the action of steam on valve-pistons. Fig. 79 shows the valve-gear and steam-cylinder in detail. The steam-chest consists of the cylinder M, valve A, and the pistons B and B′; the pistons are connected, sufficient space being allowed between them for the valve A and the steam-ports f and f′.

The valve is controlled and operated by the steel cam C acting on steel pin D, which passes through the valve into exhaust-port in which the cam is located. In addition to this mechanical operation steam is alternately admitted to and exhausted from the ends of the steam-chest by ports e and e′, operating the pistons B and B′.

OPERATION. — The pump being at rest with the valve A covering the main steam-ports f and f′, the cam C holds the valve by means of valve-pin D, so that ports e and e′ admit steam to one end of chest, and connect the other end with exhaust-port; the steam acting on valve-pistons will move valve-

pistons and valve, opening main ports f and f′, admitting steam to one end of steam-cylinder, and opening the other end to the exhaust.

If the valve occupies any other position, the main ports f and f′ will be opened for the admission and exhaust of steam; consequently it is evident that there is no dead point, and that the pump will start from any point of stroke.

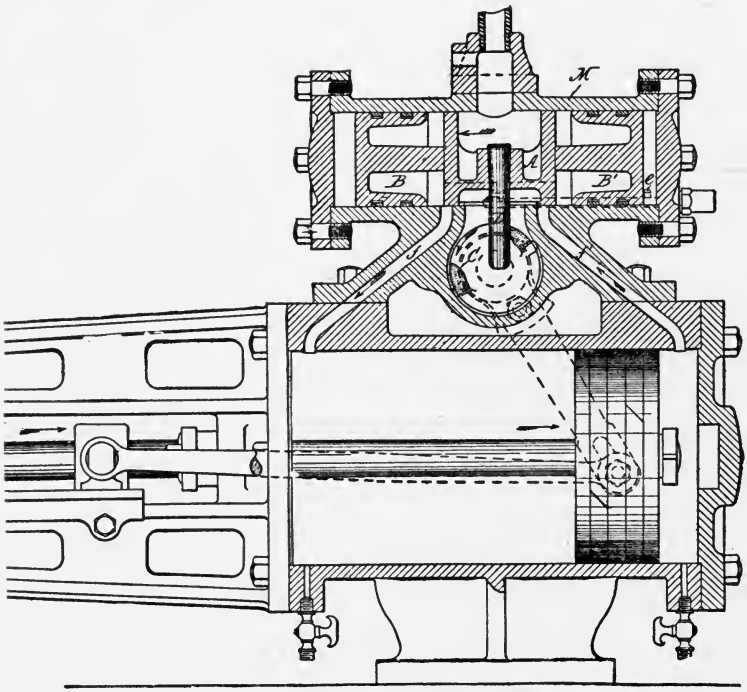


Fig. 79. The Davidson Valve Gear.

Steam being admitted to cylinder by one of the main ports, as f′ in illustration, the steam-piston, cam, valve, etc., will move in direction indicated by arrows. The first move of the cam will be to oscillate the valve preparatory to bringing it in proper position for the opening of the auxiliary steam-port e to live

steam, and e′ to exhaust, and secondly to bring the valve to its closure (mechanically) slightly before the end of the stroke of main piston (thereby causing slight cut-off and compression), and fully opening auxiliary port e to steam and e′ to exhaust. By the admission of steam to one end of the chest, the other being open to exhaust, the valve-pistons will move valve to such position as will allow the admission and exhaust of steam to and from cylinder for the return stroke.

The main valve being as much under control of the piston-rod as is the valve of an ordinary steam-engine worked by an eccentric, secures a positive action, pump being capable of starting from any position and maintaining a uniform and full stroke.

The steam-piston is absolutely prevented from striking the cylinder-heads by virtue of the mechanical valve-closure.

THE CAMERON PUMP GEAR.

Fig. 80 shows the valve-gear of the Cameron pump. There are two plain tappet-valves, one in each cylinder-head. Short valve-stems project into the cylinder; and when the piston strikes one of these the valve is driven back and opens an exhaust-passage from the corresponding end of the chest-plunger, which immediately is shifted under the action of live steam on the opposite side of the plunger-head. There is a small hole in each end of the hollow plunger; and when both tappet-valves are closed the steam, passing through these holes, leaves the plunger

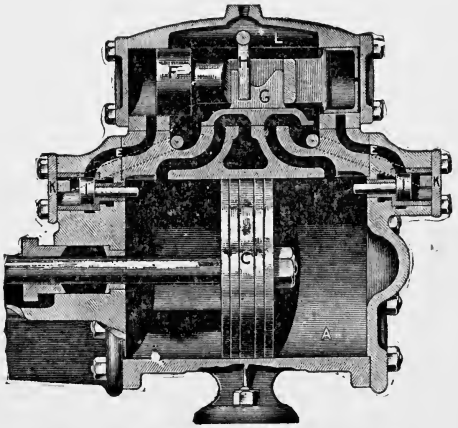


Fig. 80. The Cameron Pump Gear.

entirely surrounded by live steam, and therefore in perfect balance endwise, until the piston strikes the tappet in the opposite cylinder-head, when the valve-moving operations are repeated in the opposite direction. The space back of the tappet-valve communicates with the steam-chest through a passage, shown in dotted lines; and the valve is therefore closed by steam-pressure as soon as the piston moves back from the stem. It will be noticed that the piston closes the exhaust-passage before the end of the stroke. The confined steam forms a cushion between the piston and the cylinder-head, but a little passage is cut in the cylinder wall through which sufficient steam is admitted to start the piston on the return stroke.

The main valve, carried by the chest-plunger, is shifted in the direction the piston travels at the end of the stroke, that is, in a direction opposite to that of a common slide-valve. This valve has, therefore, two cavities, each of which alternately puts the cylinder in communication with the steam-chest and the central exhaust-port. Steam is admitted under the outer valve-face, as shown in the cut. H is a lever, by means of which the steam-chest plunger may be reversed by hand when expedient.

WORTHINGTON.

Fig. 81 shows one side of a Duplex Pump built by Henry R. Worthington.

Each individual pump actuates a rocker, which drives a valve on the pump on the other side. That is, the rocker of one pump and the valve of the other pump are constantly in gear. Each valve is, therefore, moved independently of the piston in the same pump. The mutual operation is such that one pump must always be half a stroke, or more, behind the other. At the end of the stroke the valve covers both steam-ports; but then the other pump is at mid-stroke, with its ports fully uncov-

ered, and thus the valve is always promptly carried over its dead center by the piston of the other pump.

As one piston is about half a stroke ahead of the other, and the valve-motion should be the same on both pumps, it follows that one of the rockers must have a counter-arm above its fulcrum to move the valve in the proper direction.

It will be noticed that the valve is confined between nuts on the valve-stem, and that they are not set closely against the valve; there is "lost motion." This lost motion is necessary, for otherwise the pump might not make full stroke. This is explained by the fact that the first part of the stroke is always made in less

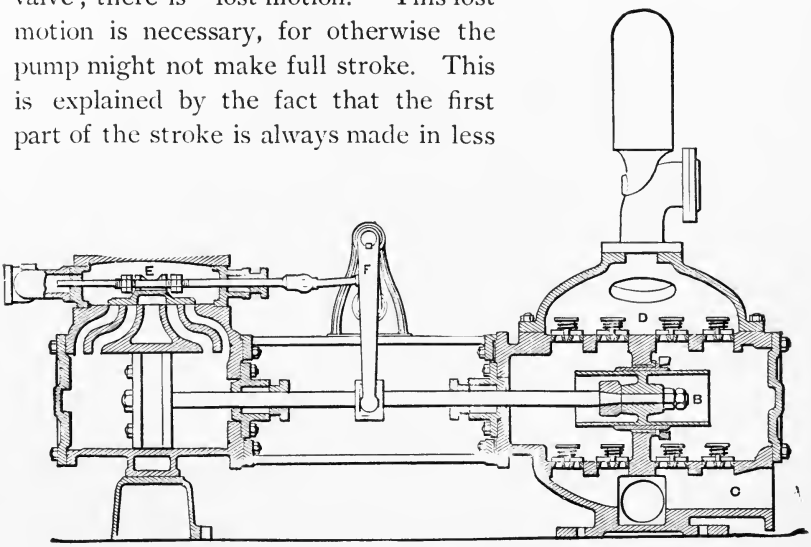


Fig. 81. The Worthington Duplex Pump.

time than that of a corresponding part later in the stroke; wherefore the mid-point, in regard to time, or the middle of the stroke-period, comes later than half-stroke. The lost motion corrects this discrepancy when the valve-motion is reversed.

A too tightly packed water piston or plunger may cause a pause at the commencement of the stroke, when the steam-ports are nearly covered, and consequently a later reversal of the valve-motion on the other side; and from this cause the working of

the pump may become somewhat irregular. The water-packing of a duplex pump should always be in good condition.

All duplex pumps have separate exhaust-ports, as shown in the figure. The inner ports are only for exhaust, and the outer ones for live steam. After the exhaust-port has been opened, there remains exhaust-steam at atmospheric pressure in the cylinder and steam-passages; this is expelled during the return stroke until the exhaust-opening is covered by the piston; the remaining steam is then compressed, and forms a steam cushion between the piston and cylinder-head. One of the exhaust-ports is always covered by the valve.

Steam cannot be used expansively in single-cylinder direct-acting pumps, because there is not enough kinetic energy stored in the water and moving parts to carry the piston to the end of its stroke; but it may be used expansively in multi-cylinder pumps, for the pressure in each individual cylinder may not vary greatly in a compound system, when the admission is continuous during the entire stroke. The result is, of course, a considerable saving in steam. The speed of a pump is regulated by throttling the steam, and this has the disadvantage that during the dwell at the end of the stroke the steam-pressure in the cylinder is apt to rise; that is, at a point where the increased pressure does no good.

In a triple compound pump the work done in the high-pressure cylinder is only a small part of the whole work; and it is, therefore, feasible to cut off the steam in this cylinder before the end of the stroke.

Fig. 82 shows, in section, the high-pressure cylinder of a "Triple Pump," built by Henry R. Worthington. It has three cylinders, connected in tandem, and it is on the Duplex plan; each set of valves is driven by a rocker, which receives its motion from the pump on the other side. The valves are all of the semi-rotary kind. They are, in fact, nothing but common D-valves on concave valve-seats. They are located under the cyl-

inders for convenience, and it is a superior arrangement for draining the cylinders, and the whole arrangement is eminently suitable for a simplified valve-gear.

There are two steam-ports and two exhaust-ports in the steam-chest, arranged as the ports in an ordinary duplex pump; but the steam and exhaust passages are united into one passage before entering the cylinder. In the steam-passage is a small

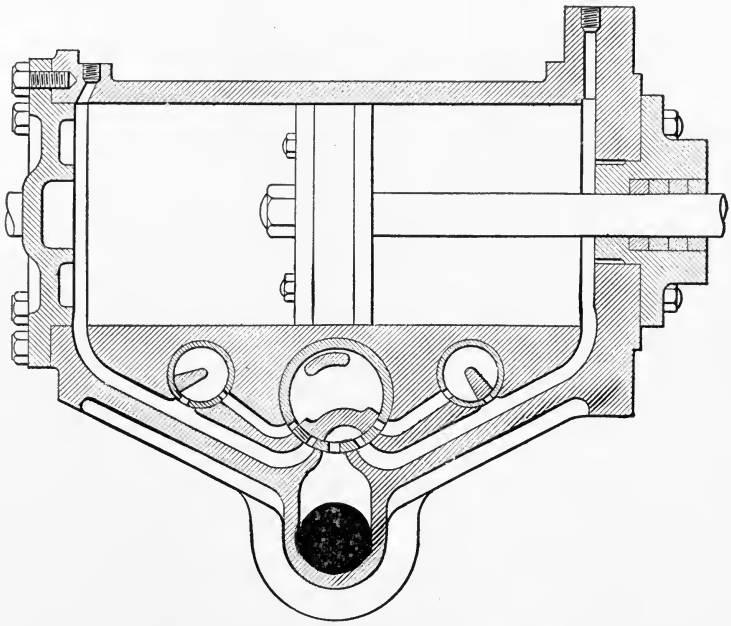


Fig. 82. The Worthington Triple Gear.

valve which cuts off the admission of steam before the end of the stroke. The point of cut-off is ordinarily fixed, but it can readily be changed by lengthening or shortening the rod connections; that is, by giving the cut-off valves more or less lap.

There are only two ports in the intermediate and low-pressure cylinders; and the opening into the low-pressure cylinder is so located that the piston covers it before the end of the stroke, and is thus cushioned on the imprisoned steam.

CHAPTER VI.

ANGULARITY OF CONNECTING-ROD AND ECCENTRIC-ROD.

THE ANGULAR MOTION OF THE CONNECTING-ROD.

THE motion of the crank-pin is both horizontal and vertical; that is, it may be dissolved in horizontal and vertical components, parallel with the center line of the engine and at right-angles to it; and it was in the first chapter of this book assumed that the horizontal components, or the projections of the crank movements on the horizontal diameter of the crank-circle, represent exactly the movements of the piston in a horizontal engine, and no account was taken of the vertical, or up-and-down motion of the crank-pin. In other words, it was assumed that the horizontal distance from the crank-pin to the cross-head pin is the same at any point of the stroke, which is not true; but the assumption is permissible, because it is sufficiently near the truth for a general investigation of the valve-action, and it greatly simplifies the construction of valve-diagrams. When the crank-pin is right on the center-line of the engine, its distance from the cross-head pin equals the length of the connecting-rod; but when the crank-end of the connecting-rod is above or below the center-line, the horizontal distance between the pins is shortened more or less, according to the inclination of the rod. The piston is, therefore, always **nearer the crank-end of the cylinder** than has been assumed, except in its two terminal positions; and in equal periods more steam is admitted during the head-end stroke, and less during the crank-end stroke. The piston moves faster in the head-end part of the cylinder and slower in the crank-end part; and when the crank is in one of its "quarter" positions, vertically

above or below the shaft, the piston has either passed its middle position in the cylinder or it has not yet reached it, according to which way it is moving. In constructing the valve-diagrams it was assumed that both crank and eccentric turned at a uniform rate, or moved through equal angles in equal time, which assumption is correct, and on this basis coincident positions of the piston and valve were determined; but, as no attention was paid to the angularity of the connecting-rod, some correction is necessary to present the piston-motion in its exact relation to the invariable time basis, and to suggest such changes of the valve and valve-motion as will cause it to operate in closer harmony with the irregularities of the piston-motion.

Hitherto one side of the piston only was considered, on the assumption that there was no difference in the piston-velocity from either cylinder-head, but now both sides of the piston will be considered conjunctively. A double-acting cylinder represents two single-acting cylinders, as it were, and it appears that the piston-velocity is not the same in both of these. The main difficulty consists in equalizing the cut-offs on both sides of the piston when steam-admission and cut-off is regulated by a single-valve or conjugate-valves; and this is the condition assumed in the following.

In a great number of engines a slight inequality of the cut-offs is of no consequence, and no attention is paid to it, or it is partly corrected by the valve-setting. No power is gained by equalizing the cut-offs, and no steam is saved by it. The "most economical point of cut-off" is a vague conception, and the variation due to the angularity of the connecting-rod is not sufficient to have any practical bearing on this point; but inequality of cut-off has unquestionably some disturbing effect on the speed of the engine, and this may be of some consequence in compound or high-expansion engines.

The angularity of the eccentric-rod has a disturbing effect on the steam-lead which may become of some consequence in

shifting eccentric-engines, and Professor Sweet's method of obviating it will be explained.

The circle in Fig. 83 represents the path of the crank-pin, and the direction of motion is shown by the arrow. The circle may also represent the time, or period of one revolution of the crank; and as the rotative speed is supposed to be uniform, any part of the circle, measured in degrees or by its center angle, will represent a proportionate period, or interval of time; and the center-line of the crank, like the hand of the clock turning on its dial, will mark fractional parts of the period of revolution, in strict conformity with its angular movements.

While the crank is turning, the valve slides on the valve-face, and is supposed to open and close the ports at the proper

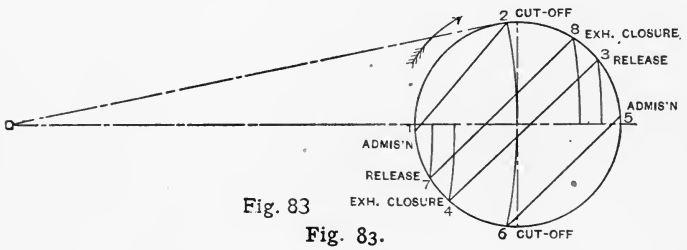


Fig. 83

Fig. 83.

moments for an **equal steam distribution** at both cylinder-ends, or at both sides of the piston; and the instant this occurs, or should occur, may be marked on the crank-circle, as shown.

The diameter of the circle in Fig. 83 may represent the stroke of the piston toward the shaft from left to right and returning in the opposite direction; and it may represent forward- and return-stroke for both cylinder-ends, or for both single-acting cylinders, as it were. Considering the head-end stroke, when the crank-pin is at point 2 the piston is at the middle of its stroke; for if the connecting-rod be swung down from this point, without moving the cross-head, it will strike the center of the circle, as shown. Therefore, if steam is to be cut off exactly at half-stroke, it must occur when the crank is

in position 2, or the valve must close the port at the moment the crank arrives at point 2; and in that case arc 1–2, marks the admission-period. Considering the crank-end, the port should be closed when the crank-pin is at point 6; and in this case arc 5–6 represents the period of admission, **which is longer than the period of admission at the head-end.** The points of admission 1 and 5 are made diametrically opposite in order to get equal lead at both ends of the cylinder.

Now the eccentric turns in unison with the crank; and if the points marked on the crank-circle be turned forward, through an angle equal to the angular advance of the eccentric,

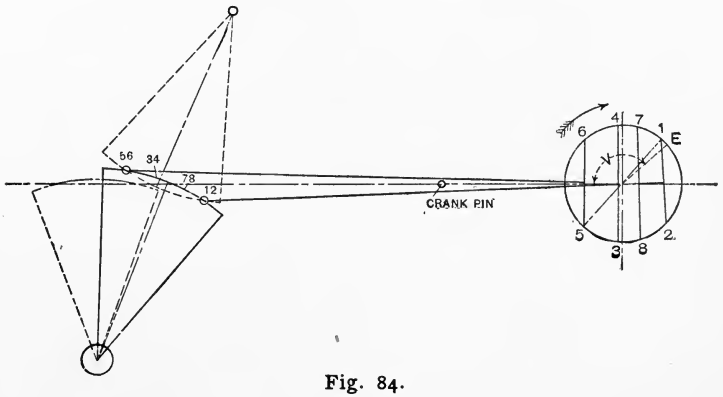


Fig. 84.

they will be in their proper positions for the eccentric-circle. Thus Fig. 84 represents the eccentric-circle. V is the angle of advance, and the figures along the circumference mark **the instants of opening and closing the ports,** as in Fig. 83, but with the eccentric-radius as a pointer.

If the valve-end of the eccentric-rod moves in a straight line which passes through the center of the shaft, the chords 1–2 and 5–6 must be parallel, as shown in the first part of the book; and which also will appear from a mere inspection of Figs. 84 and 85, if it be remembered that the valve-end of the rod must be at the same point when the valve opens and closes

the port; but, referring to Fig. 83, it will be observed that if points 2 and 6 are fixed, chords 1–2 and 5–6 cannot be parallel without changing the locations of points 1 and 5 in opposite directions. It is, therefore, **not possible to equalize the cut-offs with the ordinary eccentric-rod motion without introducing unequal leads.**

Earlier cut-off can be obtained in the head-end stroke by increasing the head-end lap, and later cut-off may be obtained in the crank-end stroke by decreasing the crank-end lap. If the valve admits steam over the outside edges, both these changes may be made by lengthening the valve-rod; and shortening it has the same effect, if steam is admitted over the inside port-edges, or if the valve is moved by a counter-arm on the rocker; but, in any event, such change involves less lead at the head-end and more at the crank-end. If the valve has ample lead, a little change either way will not be of much consequence; but if there is little or no lead or negative lead, unequal steam-laps are more objectionable; and, as the port-opening equals half the travel, minus the lap, the opening for early cut-offs may become unduly restricted at one end of the cylinder. This, however, may under certain conditions be partly avoided by using a short eccentric-rod, as will presently be explained.

THE ANGULAR MOTION OF THE ECCENTRIC–ROD.

The up-and-down motion of an eccentric of a horizontal engine draws the eccentric-rod towards the shaft, as shown in Fig. 85, where the rod is represented disproportionally short for the sake of clearness. To correct this, the immediate valve connections must be lengthened if the valve moves in the same direction relative to the crank-shaft as does the eccentric-rod pin, and shortened if the motion is reversed by a counter-arm. In other words, the head- and crank-end laps are made unequal.

The angular vibration of the eccentric-rod gives a quicker motion to the valve at one end of its travel and a slower motion

at the other end; and the period of admission is, therefore, **under certain conditions,** shortened at the head-end and lengthened at the crank-end **without changing the laps.** The result is more equal cut-offs with **equal port-openings;** but if the admission periods must be diametrically opposite, the inequality of these periods must necessarily make the leads unequal.

In order to obtain results as here stated, the steam must either be admitted over the inside port-edges, or else the valve must be moved by a counter-arm rocker; **otherwise the inequality of laps and port-openings will be increased instead of diminished.**

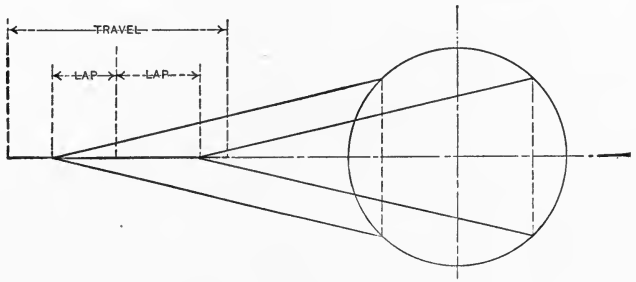


Fig. 85.

The length of the eccentric-rod, compared with the throw of the eccentric, is generally too great to make the influence of its angular vibration of much consequence; but if the facts here mentioned are fully understood, they may occasionally be used to advantage by a discriminating designer.

If the valve-end of the eccentric-rod is guided by a common rocker-arm, it may become slightly elevated at the middle of its travel, which has a quite inappreciable effect in the direction of later release and compression if the engine is "running over," and a depression towards the end of its travel has the opposite effect on the lead and cut-off, but the effect in either case is **practically nil.**

STEAM- AND EXHAUST-LAPS.

As the diameter of the crank-circle, Fig. 83, represents the forward stroke and the return-stroke for both ends of the cylinder, points of cut-off, release, and exhaust-closure may be located on this diameter at equal distances from the ends; and by striking arcs with radius equal to the length of the connecting-rod, as shown, the corresponding crank-positions are located.

Points 1, 2 and 3, 4 mark admission and cut-off, release, and exhaust-closure positions of the crank-pin for the head-side of the piston, and 5, 6 and 7, 8 mark corresponding positions for the crank-side. Ordinarily all the chords connecting each pair of these points **must be parallel,** and to suit this requirement, the points must be rearranged, according to the best judgment of the designer.

The distances from the center of the circle to the chords, measured by the proper scale, give the required laps; and it will be noticed that release and compression may be very nearly equalized for both cylinder-ends by suitably proportioned exhaust-laps. The exhaust-lap for the crank-end becomes larger than that for the head-end, and the only objection to this is the reduction of port-opening at the crank-end. If the exhaust-laps are made equal, it is impossible, with the ordinary valve-gear, to make the compression curves on the indicator card equal without greatly disturbing the lead and cut-off.

EQUALIZING BOTH LEAD AND CUT-OFF.

If the cut-offs are to be equal on both sides of the piston **without changing the leads** the periods of admission cannot be diametrically opposite. Any period may be changed either way by turning the eccentric on the shaft; but in that event all the periods will be changed conjunctively. Instead of turning the eccentric on the shaft, the eccentric-rod may be turned; for the eccentric-action is in the direction of the rod; and if the rod is

turned in a different direction, the points marking opening and closing of the ports will be turned the same amount in the same direction on both the eccentric and crank-circle; and if the valve-end of the rod receives a reciprocating angular motion relative to the crank-shaft, there will be a corresponding angular displacement of the conjugate points,* according to the instantaneous angular displacement of the rod; and, therefore, **by suitably guiding the end of the eccentric-rod, the admission-periods can be properly timed for equal leads and equal cut-offs** on both sides of the piston.

The mean direction of the eccentric-rod for any period of open or closed port is represented by a straight line passing through the center of the corresponding arc on the eccentric circle and at right angles to the chord. If the valve-end of the eccentric-rod be guided by an inclined rocker-arm, as shown in Fig. 84, the mean direction of the rod for the admission period 1–2 will be different from its mean direction for the admission-period 5–6, as indicated in the figure; it being understood that when the eccentric is at either point 1 or 2, the pin is at point 12, and when the eccentric is at 5 or 6 the pin is at point 56 — these being the coincident positions of the eccentric-rod pin when the steam-edges of valve and port come together at the head- and crank-end respectively.

Now, if by variation of the mean inclination of the rod for points 12 and 56 respectively, the angle of variation equals the angle of convergence of the admission-chords, 1–2 and 5–6, in Fig. 83, the cut-off may be fixed at exactly half the stroke either way by giving the valve proper laps, and without disturbing the leads. The laps are determined by the distance from the center of the rocker-arm arc to points 12 and 56 respectively; and the port-openings are determined by the distance of these points from the ends of the arc.

If the points on the eccentric-circle are correctly spaced,

* Points 1, 2 and 3, 4, etc., are for obvious reasons called conjugate points.

points 12 and 56 may be located by striking arcs with a radius equal to the length of the eccentric-rod from the corresponding points on the circle; and for the equalization of lead and cut-off, the end of the rod may be guided in any manner whatever if it passes through points 12 and 56; but the rocker-arm affords the only satisfactory means of accomplishing this. The rocker-fulcrum may be located above the eccentric-rod; but this would make the port-openings unequal, as shown in Fig. 84, while in the other position the port-openings may be equalized by using a rocker-arm of the proper length. The exhaust-periods will be advanced a trifle on account of the upward curvature of the rocker-arc; that is, they will come a trifle later in the stroke, which may be an advantage rather than otherwise. The curvature of the arc has practically no effect on the equalization of release and compression, which depends on the correct proportioning of the exhaust-laps.

THE INCLINED ROCKER FOR SHIFTING-ECCENTRIC ENGINES.

When the inclined rocker is applied to variable cut-off engines it cannot equalize the cut-offs at all points of the stroke; for the equalization depends on the angle between the two mean directions of the eccentric-rod for the head-end and crank-end admission-periods, as shown in Fig. 84. As the laps of the valve are unchangeable, the distance between points 12 and 56 is practically constant for all cut-offs; and the rocker-arm can, therefore, only equalize the cut-offs for two or four points on the crank-circle, which give the proper convergency or mutual inclination of the admission-chords. This convergency is greatest for the mid-stroke cut-off, and it decreases by earlier and later cut-offs, and becomes practically nil at the beginning of the stroke. If, for example, the rocker-fulcrum be so located as to equalize the cut-offs at one-fourth of the stroke, the earlier cut-offs will be too early at the head-end and too late at the

crank-end; and an ordinary rocker-arm will give very nearly equal cut-offs near the beginning of the stroke.

EQUALIZING THE LEAD IN SHIFTING-ECCENTRIC ENGINES.

The angular vibration of the eccentric-rod draws the valve towards one end of the steam-chest, and this can be provided for by making one lap a little longer than the other; but if the throw of the eccentric varies, as when it is shifted across the shaft, the lead cannot be equalized in this manner for more than one proportion of cut-off, but nearly equal leads at both cylinder-ends may be obtained for the whole range of cut-off by means of the **inclined rocker,** as will now be explained.

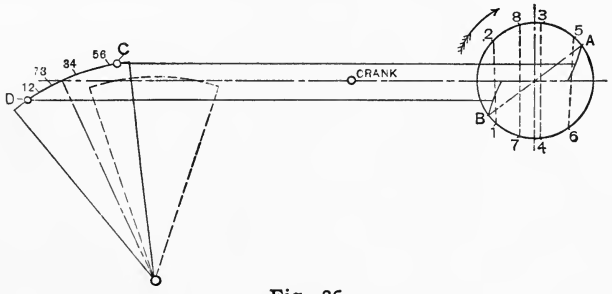


Fig. 86.

If steam is admitted over the inside port-edges, the eccentric must be located on the crank-side of the shaft; and in that case Fig. 86 represents the eccentric-circle, and the rocker will, in its middle position, be leaning toward the cylinder-end, as shown. Let the center of the eccentric be at B and A at the commencement of the head-end stroke and the crank-end stroke respectively, and let the annexed short curves represent the path of the center of the eccentric, when it is moved across the shaft; then if the corresponding positions of the eccentric-rod pins are at D and C respectively, and if lines from these points to the middle points of curves B and A are parallel, it will be seen that if the rod is swung from the outer to the

inner end of the curved path in either position, points D and C will be shifted very nearly the same amount, in opposite directions, and thus **nearly equal lead may be obtained for both ends of the cylinder, irrespective of the points of cut-off.*** It should be observed that **the valve-end of the eccentric-rod must be angularly advanced, relative to the crank-shaft while moving toward the shaft;** and in order that, under this condition, the cut-offs may be equalized for some intermediate points in the stroke, it is necessary that **the admission-period for the head-end be on the rocker-side of the eccentric.** The valve must, there-

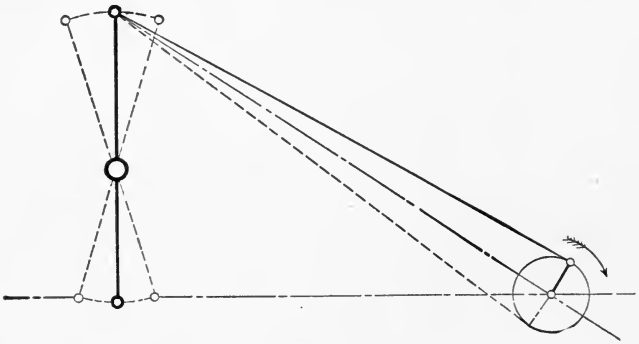


Fig. 87.

fore, **either admit steam over the inside edges, or else the motion from the eccentric must be reversed by means of a counter-arm on the rocker.** By such arrangement the cut-offs may become more equal than with the ordinary rocker; but if the crank-end admission-period is on the rocker-side of the shaft the inequality of the cut-offs will be greater than when a common rocker is used. Also note that equal port-openings cannot be had unless **the rocker-fulcrum is behind the rocker-arm, in the direction of rotation of the eccentric;** that is, if we imagine the rocker swung around the shaft.

In the Straight-Line engine steam is admitted over outside

* Professor Sweet explained this to the writer in 1882 by drawing seven chalk-lines on a blackboard.

port-edges, and the form of the rocker used on some of these engines is indicated in Fig. 87.*

UNEQUAL LAPS FOR A VARIABLE CUT-OFF ENGINE.

It is questionable whether any attempt should be made to equalize the cut-offs in shifting-eccentric engines by varying the laps without the inclined rocker; for any considerable dissimilarity of the laps will have a decidedly bad effect on the early cut-offs and early leads, on account of the short travel and small port-opening; but the release and compression may be equalized for both sides of the piston, to a great extent, by making the head-end exhaust-lap smaller and the crank-end lap longer. In the early cut-offs, the valve opens and closes for the exhaust nearer the middle of the piston-stroke, and the greater angularity of the connecting-rod at these points neutralizes, to some extent, the effect of reduced travel as far as the unequal laps are concerned; and as the smallest exhaust-opening is never much less than the amount of steam-lap, the conditions are altogether favorable for an equalization of the exhaust-action by means of the exhaust-laps.

The graphical methods used or suggested in connection with the subject-matter of this chapter are useful in a wider sense, as illustrative of a simple and rational method for the **exact** determination of the steam distribution in the cylinder, the lapse of the valve, and the port-openings a method which is generally applicable, and which involves few artifices and no curve-tracing, and therefore should commend itself to the practical designer.

* The inclined rocker was discussed by Mr. F. A. Halsey in the *American Machinist* for February 28 and March 14, 1889.

HALLIDIE LIBRARY OF THE UNIVERSITY OF CALIFORNIA

INDEX.

[Illustrations are indicated by an asterisk (*).]

PAGE

Admission 9, 10
Period of, 5*, 8, 9, 10, 11, 12*, 15, 16, 21, 131, 135, 138
Point of 9, 11, 12*
Allen, John F. 75
Allen Locomotive Valve, the 35*
Allfree Valve Gear, the 57, 58*
Angle of Advance, 5*, 10, 11, 12*, 14*, 15, 16, 21
Angularity of Connecting-rod, 11, 128, 129, 130*
Of Eccentric-rod 6, 129, 132, 133*

Ball, Frank H. 41
Telescopic Valve, the 41*
Balanced Area 43*, 45*
Balancing a Common D-valve . 42, 43*, 44, 45*
Balance, the Thomas 45*
Balanced Valve with Independent Cut-off 101*
Begtrup's Eccentric 97, 99*
Blake Single Pump, the 117*
Buckeye Flat Valve, the 103*
Round Valve, the. 106*
Valve Gear, the 102, 104*
Vibrating Cut-off 111, 112*, 113*

Cameron Pump-gear, the 123*
Clark, D. K. 27
Clearance, 27, 28, 35, 52, 53*, 54, 63*, 64*, 80, 111*
Combination Valve, Limitations of the, 16, 17
Compound Rocker, the Buckeye 70*
Valves . . . 50, 51*, 52, 53*, 54, 55, 56*
Compression 11, 12*, 17, 27
Condenser, Effect of 27, 43
Connecting-rod, Angularity of, 11, 128, 129, 130*
Cooling Surface 34, 35, 76
Corliss, George 31
Corliss Gear, Limitations of 65, 66
Setting of the 72
Corliss Valves 1, 62*, 63*
Dimensions of 71
Motion of 62*, 63*, 64, 65

PAGE

Creeping Steam 43
Cut-off 10, 11, 12*, 16, 17
Eccentric Journal on Engine Shaft, 90, 91, 92
Equalized 132
Gonzenbach, the 81*
Independent 80
Meyer's 85*
Point of 5*, 11, 12*, 16
Range of 25
Single Valve 23
Square 22, 31
Unequality of 129
Varied by Rotating Eccentric, 90*, 93*, 96*
Cut-off Valves, 80, 81*, 85*, 90*, 93*, 96*, 101*, 103*, 106*, 109*, 110*, 112*, 113*
On Back of Main Valve 83
On an Anchor Plate 83
On a Stationary Valve Seat 81
Simple Form of 81, 82*
With Constant Travel on Main Valve, 99, 101*, 103*, 106*, 112*
Worked by Means of a Rocking Link, 106, 107*
Cylinder, Double Acting 4, 229
Single Acting 4, 50*, 53*

Dash-pot Action 65, 66
Dead Center 9, 11
Setting Engine on 32
Davidson's Pump Gear 121, 122*
Dean Brothers' Single Pump Gear . 119, 120*
Double Ported Valves . . 34*, 39*, 41*, 63*
Duplex Pump 125*
D-valve 1, 2*, 4, 7, 8, 18, 41, 96
Diagram 12*
Cylindrical 78*, 127*
Limitations of the 16, 17
On Pumps, the 115
Double-Acting Engine 4, 8

Eccentric 5, 9, 11
Eccentricity of 6

PAGE

Eccentric, At Half-Throw 7
Exhaust 8, 70
Begtrup's 97, 99*
Radius of 6
Rod, Angularity of . . . 6, 129, 132, 133*
Single 17
Shiftable 16, 21
Steam 8, 69
Throw of 6, 16
Rotative on Shaft . . 83, 90, 103, 112, 113
Exhaust Cavity 2*, 8, 17, 20, 41
Eccentric 8, 70
Clearance 4
Closure 12, 17
Edge 17
Lap 2*, 3, 7, 8, 9, 17
Negative 3, 56*
Lead 10, 12
Line 19
Opening 8, 16, 38
Passage 18
Period 5*, 8, 9, 12*, 15, 16
Port 3, 7, 8, 62*, 63*
Valve 7, 21, 37, 111*

Face of Valve 2
Free Expansion 31
Four-valve Systems 61
Friction of Steam 19

Gonzenbach Cut-off, the 81*
Valve Diagram of 83*
Governor, Shaft, 37, 50, 53, 92, 93, 99, 109, 110, 112
Vertical 107
Gridiron Valves with Independent Cut-off, 96
Gridiron Valves . 75, 76*, 96* 109*, 112,* 113*

Halsey, F. A. 139
Hill, Edward K. 76
Hill Valves, the 76*

Ideal Engine, Valves of the . . . 47*, 48*
Independent Cut-off 80
On Four-valve Engines 108
Improved Slide-valves 34
Inclined Rocker for Shifting-Eccentric, 136, 137*, 138*
Indicator Diagram, Theoretical . . . 14,* 24
Inside Lap 3
Intervening Steam Pressure 44

Knowles' Pump-gear 121

Lap 2*, 5*, 7, 9, 11, 15, 16
On Separate Exhaust Valve 23

PAGE

Lap, Travel and Port-Opening . . . 24, 25
Steam and Exhaust 134
Unequal 133, 134, 135, 139
Lapless Valves 11
Leakage Loss 32
Lead 10, 12*, 16, 21*
Angle 26
Negative 10, 27
Notes About 25
Variable 21*, 26, 29*, 31
Limitations of the Combination Valve . . 16
Of the Corliss Gear 65, 66
Links 29*, 72, 107*
Locomotive Valves, Motion of . . . 28, 29*
Low Receiver Pressure 43

McEwen Valve, The 39*
McIntosh & Seymour Valves, The . 108, 109*
Meyer Cut-Off, The 85*
Limitations of 89
Multiporting 31

Negative Lap on Cut-Off Valves, 82*, 86*, 87, 99*, 100*, 101*, 104*
Negative Exhaust-Lap 3, 56*
Lead 10, 27

Outside-Lap 3
Oscillating Valves 1

Piston-Valves 1, 46*, 47*
With Independent Cut-Off . . 105, 106*
Piston Velocity 22, 25*
Period of Revolution 9
Of Steam Distribution 11
Port 2*, 18, 19*, 20
Area of 19
Contraction of 18, 19,* 43*
Formula for 19
Port-Opening . . . 7, 9, 15, 17, 19,* 22, 23, 25*
Equality of 138
Initial 10
Variation of 24, 25*
Porter Allen Engine, Valves of the . . . 72*
Pressure Relieving Device . . . 42, 43*, 45*
Plate 36*, 38, 73*, 74
Exhaust 38
Pumps, Slide-Valve on 115
Blake, the 117*
Cameron, the 123*
Davidson, the 121, 122*
Dean Brothers' 119, 120*
Knowles' 121
Worthington, the 124, 125*
Triple Compound 126, 127*

PAGE

Receiver Pressure, High 51
Low 43
Relation Between Lap and Travel . . 15, 21*
Release, Point of 12, 23, 24
Early 24
And Compression, Constant, 21*, 23, 38, 57, 80
Relieved Area 43
Reversal of Crank Motion 15
Revolution, Period of 9, 16
Rice & Sargent, Valves Designed by . . 64*
Richardson Balanced Valve, the 44
Rider, A. K. 90
Cut-Off, the 89, 90
Rocker, Compound 103, 104*
Counter-Arm 30, 133, 138*
Inclined 135, 136, 137*, 138*
Robinson, S. W. 43
Russell Engine, Valves of the . . . 110*, 111*

Semi-Rotary Valves . 1, 62*, 63*, 64*, 78*
Single-Valve Gear, Allfree, the . . . 57, 58*
Single-Acting Cylinders 4, 50*, 53*
Slide-Valve, the Common 1, 2*
Cylindrical 1, 62*, 63*, 64*, 78*
Simplest Form of 4, 5*, 8
Spring-Rings 47, 48, 106*
Steam, Creeping 43, 45
Admission 7
Friction of 19
Dry, Wet, or Superheated 19
Eccentric 8
Lap 2*, 3, 7, 11, 12*
Lead 10
Line 19, 31
Port 2*, 3, 18
Passages 18, 32
Ports, Supplementary, 17, 32, 35*, 36*, 46*
And Exhaust Edges, Transposition of, 38
Velocity of 19
Valve 7
Straight Line Balanced Valve, the . . . 36*
Variations of 38, 39*
Sweet, John E. 15, 130, 138
Sweet's Valve Diagram 14*, 15, 21*

Telescopic Valve 41*
Thomas, W. J. 44
Thomas Balance, the 44, 45*
Thomson, J. W. 102

PAGE

Throw of Eccentric 6, 12*, 16
Travel and Lap 21*, 22
of Valve 9, 12*, 15, 17

Unequal Laps for Variable Cut-Off Engines, 139

Valve Dimensions 20
Face of 2
Seat of 2
Motion in Relation to Piston Motion . 9
Without Lap 11
Diagrams, Primitive 12*
Sweet's 14*, 21*
For Corliss Engines . . . 67*, 70*
Simple Cut-Off 82*
For the Gonzenbach Cut-Off . . 83*
For Meyer's Cut-Off 86*
For Rotative Adjustment, 91*, 92*, 94*
For Link Connection 107*
Combined Steam and Exhaust . . 16* 17*
For High-Speed Engines, 16, 35*, 36,* 39*, 41*, 43*, 45*, 46*, 47*, 48*, 50*, 51*, 53*, 56*, 58*
Flat Balanced 36*, 39*, 73*
Leakage of 32
Separate for Steam and Exhaust, 21, 61, 63*, 73*, 76*
Unbalanced 17
Cut-Off 80
Expansion 80
Gridiron 75, 76*, 96*, 109*, 113*
Of the Straight-Line Type 37, 39*
Compound 50, 51*, 53, 56*
Designed by Messrs. Rice & Sargent, 63, 64*
Variable Valve Travel 15
Vauclain Valve, the 54, 56*
Velocity of Steam 19
Piston 25*

Westinghouse Standard Valve, The, 49, 50*
Compound Valve, the 53*
Wheelock Valves, the 78*, 79
Wiredrawing 31, 37
Worthington, Henry R. 116
Duplex Pump, the 124, 125*
Triple-Pump Gear, the 127*

Zeuner, Dr. 88

CLASSIFIED CATALOGUE OF

BOOKS ON STEAM, STEAM ENGINES, Etc.

FOR SALE BY

D. VAN NOSTRAND COMPANY,

23 MURRAY AND 27 WARREN STS., NEW YORK.

BOILERS.

Barr. Practical Treatise on High Pressure Steam Boilers, including Results of Recent Experimental Tests of Boiler Material, etc. 8vo. Illustrated. Indianapolis, 1893. $3.00

Barrus. Boiler Tests: Embracing the results of one hundred and thirty-seven evaporative tests, made on seventy-one boilers, conducted by the author. 8vo. New York, 1900. $3.00

Christie. Chimney Design and Theory. A book for Engineers and Architects, containing all data relative to Chimney Designing. Illustrated with numerous diagrams and half-tone cuts of many famous chimneys. 8vo, cloth. Illustrated. New York, 1899. $3.00

Courtney. The Boiler Maker's Assistant in Drawing, Templating, and Calculating Boiler Work and Tank Work, with rules for the Evaporative Power and the Horse Power of Steam Boilers, and the Proportions of Safety Valves, and Useful Tables of Rivet Joints of Circles, Weights of Metals, etc. Revised and edited by D. K. Clark, C.E. Illustrated. London, 1901. (Weale's Series.) $0.80

The Boiler Maker's Ready Reckoner. With examples of Practical Geometry and Templating, for the Use of Platers' Smiths, and Riveters. Revised and edited by D. K. Clark. 4th Edition. London, 1899. (Weale's Series.) $1.60

Christie, W. Wallace. Chimney Design and Theory. A Book for Engineers and Architects, with numerous half-tone illustrations and plates of famous chimneys. 2d edition, revised. 8vo, cloth. $3.00

Foley, Nelson. The Mechanical Engineer's Reference Book for Machine and Boiler Construction, in two parts. Part I., General Engineering Data. Part II., Boiler Construction. With 51 Plates and numerous illustrations specially drawn for this work. Folio, half mor. London, 1895. $25.00

Horner. Plating and Boiler Making. A Practical Handbook for Workshop Operation, including an Appendix of tables by A Foreman Pattern Maker. 338 illustrations. 12mo. London, 1895. £3.00

Hutton. Steam Boiler Construction: A Practical Handbook for Engineers, Boiler Makers, and Steam Users. With upwards of 300 illustrations. 3d edition. 8vo. London, 1898. $6.00

Munro. Steam Boilers: Their Defects, Management, and Construction. 3d edition enlarged, with numerous illustrations and tables. 12mo. London, 1889. $1.50

Roper. The Steam Boiler: Its Care and Management. With instructions for increasing the Efficiency and Economy, and insuring the Durability and Longevity of all classes of Steam Boilers, Stationary, Locomotive, Marine, and Portable. With Hints and Suggestions and Advice to Engineers, Firemen, and Owners of Steam Boilers. 4th edition, revised. 12mo, tuck, mor. Philadelphia, 1897. $2.00

—— **Use and Abuse of the Steam Boiler.** Illustrated. 12th edition. 12mo, mor. tucks. Philadelphia, 1897. $2.00

Rose. Steam Boilers. A Practical Treatise on Boiler Construction and Examination. For the Use of Practical Boiler Makers, Boiler Users, and Inspectors, and embracing in plain figures all the calculations necessary in designing and classifying Steam Boilers. 73 engravings. 8vo. Philadelphia, 1897. $2.50

Rowan. On Boiler Incrustation and Corrosion. New edition, revised and enlarged by F. E. Idell. 16mo, boards. New York, 1895. $0.50

Sexton. Pocket Book for Boiler Makers and Steam Users, comprising a variety of useful information for Employer and Workman, Government Inspectors, Board of Trade Surveyors, Engineers in charge of Works and Slips, Foremen of Manufactories, and the General Steam-Using Public. 4th edition, revised and enlarged. 32mo, roan. London, 1895. $2.00

Stromeyer. Marine Boiler Management and Construction. Being a Treatise on Boiler Troubles and Repairs, Corrosions, Fuels, and Heat. On the Properties of Iron and Steel, on Boiler Mechanics, Workshop Practices, and Boiler Designs. 2d edition. 8vo. London, 1901. $4.00

Thurston. Manual of Steam Boilers: Their Designs, Construction, and Operation. For Technical Schools and Engineers. 183 engravings in text. 7th edition. 8vo. New York, 1901. $5.00

—— **A Handbook of Engine and Boiler Trials,** and of the Indicator and Prony Brake, for Engineers and Technical Schools. 4th edition, revised. Illustrated. 8vo. New York, 1897. $5.00

Watson, E. P. Small Engines and Boilers. A Manual of Concise and Specific Directions for the Construction of Small Steam Engines and Boilers of Modern Types from five Horse-power down to model sizes. 12mo, cloth. Illustrated. $1.25

Traill. Boilers: Their Construction and Strength. A Handbook of Rules, Formulæ, Tables, etc., relative to Material, Scantlings, and Pressures, Safety Valves, Springs, Fittings and Mountings, etc. For use of Engineers, Surveyors, Draughtsmen, Boiler Makers, and Steam Users. With illustrations. 3d edition, 12mo, mor. London, 1896. $5.00

Triplex. Marine Boilers. A Treatise on the Causes and Prevention of their Priming, with Remarks on their General Management. Illustrated. 12mo. Sunderland, 1899. $2.00

Watson. Small Engines and Boilers. A Manual of Concise and Specific Directions for the Construction of Small Steam Engines and Boilers of Modern Types from five horse-power down to model sizes. 12mo, cloth. Illustrated with numerous diagrams and half-tone cuts. New York, 1899. $1.25

The intention of the author in writing this work has been to furnish specific directions and correct dimensioned plans for small engines and boilers.

Wilson. A Treatise on Steam Boilers: Their Strength, Construction, and Economical Working. Enlarged and illustrated from the Fifth English edition by J. T. Flather. 12mo. New York, 1897. $2.50

——**Boiler and Factory Chimneys:** Their Draught Power and Stability. 4th edition, 12mo. London, 1899. $1.50

FUELS.

Abbott. Treatise on Fuel. Founded on the original Treatise of Sir W. Siemens. Illustrated. 16mo. New York, 1891. $0.50

Barr. A Catechism on the Combustion of Coal and the Prevention of Smoke: a Practical Treatise for Engineers, Firemen, and others interested in Fuel Economy, and the Suppression of Smoke from Stationary Steam Boiler Furnaces, and from Locomotives. 12mo, cloth. Illustrated. New York, 1900. $1.50

Clark and Williams. Fuel: Its Combustion and Economy, consisting of Abridgments of Treatise on the Combustion of Coal and the Economy of Fuel. With extensive additions in recent practice in the Combustion and Economy of Fuel, Coal, Coke, Wood, Peat, Petroleum, etc. 4th edition. 12mo. London, 1891. $1.50

Hodgetts. Liquid Fuel for Mechanical and Industrial Purposes. Illustrated. 8vo. London, 1890. $2.50

Phillips. Fuels: Solid, Liquid, and Gaseous; their Analysis and Valuation. For the use of Chemists and Engineers, 12mo. London, 1896. $0.80

Sexton, A. H. Fuels and Refractory Materials. 8vo. Cloth. London, 1897. $2.00

Williams. Fuel: Its Combustion and Economy. Consisting of an Abridgment of "A Treatise on the Combustion of Coal and the Prevention of Smoke." With extensive additions by D. Kinnear Clark. 4th edition. London, 1891. $1.50

GAS ENGINES.

Clerk. The Theory of the Gas Engine. 2d edition, with Additional Matter edited by F. E. Idell. 16mo. New York, 1891. $0.50

—— **The Gas Engine.** History and Practical Working. With 100 illustrations. 8th edition, revised. 12mo. New York, 1899. $4.00

Donkin. A Text-Book on Gas, Oil, and Air Engines: or Internal Combustion Motors without Boiler. 154 illustrations. 3d edition, revised and largely rewritten. 8vo. London, 1900. $7.00

Goodeve. On Gas Engines: with Appendix describing a Recent Engine with Tube Igniter. 12mo. London, 1887. $1.00

Hiscox, Gardner D. Gas, Gasoline and Oil Vapor Engines. Their Theory and Power. 3d edition, revised and enlarged. 8vo. cloth. Illustrated. New York, 1900. $2.50

Lockert, Louis. Petroleum Motor-Cars. 12mo, cloth. $1.50

HEAT.—THERMODYNAMICS.

Anderson. On the Conversion of Heat into Work. A Practical Handbook on Heat Engines. 4th edition. Illustrated. 12mo. London, 1901. $2.25

Box. Treatise on Heat as Applied to the Useful Arts, for the use of Engineers, Architects, etc. 8th edition. 12mo. London, 1895. $5.00

McCulloch. Elementary Treatise on the Mechanical Theory of Heat and its application to Air and Steam Engine. 8vo. New York, 1876. $3.50

Maxwell. Theory of Heat. New edition, with Corrections and Additions by Lord Rayleigh, Sec. R. S. Illustrated. 12mo. New York, 1897. $1.50

Peabody. Thermodynamics of the Steam Engine and other Heat Engines. 4th edition, 8vo. New York, 1901. $5.00

Rontgen. The Principles of Thermodynamics. With special Applications to Hot Air, Gas, and Steam Engines. With additions from Professors Verdet, Zeuner, and Pernolet. Translated, newly and thoroughly revised and enlarged by Professor A. Jay Du Bois. 732 pages. 3d edition. 8vo. New York, 1896. $5.00

Tyndall. Heat considered as a mode of Motion. 6th edition. 12mo. New York, 1890. $2.50

Williams. On Heat and Steam: embracing New Views of Evaporization, Condensation, and Expansion. Illus. 8vo. Philadelphia, 1882. $2.50

HOISTING MACHINERY.

Colyer. Hydraulic, Steam and Hand Power-Lifting and Pressing Machinery. 72 large plates. 8vo. London, 1892. $10.00

Glynn. Treatise on the Construction of Cranes and other Hoisting Machinery. 7th edition. Illustrated. London, 1887. $0.60

Marks. Notes on the Construction of Cranes and Lifting Machinery. 12mo. London, 1899. New and enlarged edition. *net*, 1.50

Towne. A Treatise on Cranes, descriptive particularly of those designed and built by the Yale and Towne Manufacturing Company, owning and operating the Western Crane Company, including also a description of light hoisting machinery as built by the same makers. 8vo. New York, 1883. $1.00

Weisbach and Hermann. The Mechanics of Hoisting Machinery, including Accumulators, Excavators, and Pile-drivers. A Text-book for Technical Schools and a guide for Practical Engineers. Authorized translation from the second German edition by Karl P. Dahlstrom. 177 illustrations. 8vo. New York, 1893. $3.75

ICE-MAKING MACHINES.

Dixon. Manual of Ice-Making and Refrigerating Machines. A Treatise on the Theory and Practice of Cold-Production by Mechanical Means. 16mo. St Louis, 1894. $1.00

Leask. Refrigerating Machinery. Its Principles and Management. With numerous illustrations. 2d edition, revised. 8vo. London, 1901. $2.00

Ledoux. Ice-Making Machines: the Theory of the Action of the Various Forms of Cold-producing or so-called Ice-Machines. Translated from the French. 248 pages and numerous Tables. 16mo. New York, 1897. $0.50

Redwood. Theoretical and Practical Ammonia Refrigeration. A Practical Handbook for the use of those in charge of Refrigerating Plants. Illustrated with numerous Tables. 12mo. New York, 1896. $1.00

Wallis-Tayler. Refrigeration and Cold Storage; being a complete Practical Treatise on the Art and Science of Refrigeration. 8vo, cloth. Illustrated. *net*, $4.50

INDICATORS.

Bacon. Treatise on the Richards Steam Engine Indicator. With a Supplement, describing the latest Improvements in the Instruments for Taking, Measuring, and Computing Diagrams. Also an Appendix, containing Useful Formulas and Rules for Engineers. 23 diagrams. 4th edition. 16mo, flex. New York, 1883. $1.00

Ellison. Practical Applications of the Indicator. With reference to the Adjustment of Valve Gear on all Styles of Engines. 2d edition. 8vo. 100 engravings. Chicago, 1897. $2.00

Hemenway. Indicator Practice and Steam Engine Economy. With Plain Directions for Attaching the Indicator, Taking Diagrams, Computing the Horse-Power, Drawing the Theoretical Curve, Calculating Steam Consumption, Determining Economy, Locating Derangement of Valves, and making all desired deductions; also, Tables required in making the necessary computations, and an Outline of Current Practice in Testing Steam Engines and Boilers. 6th edition. 12mo. New York, 1899. $2.00

Le Van. The Steam Engine Indicator and its Use. A Guide to Practical Working Engineers for greater economy, and the better Working of Steam Engines. 18mo, boards. New York, 1900. $0.50

——**The Steam Engine and the Indicator:** Their Origin and Progressive Development, including the most recent examples of Steam and Gas Motors, together with the Indicator, its Principles, its Utility, and its Application. Illustrated by 205 engravings, chiefly of Indicator-cards. 8vo. Philadelphia, 1890. $4.00

Porter. A Treatise on the Richards Steam Engine Indicator, and the Development and Application of Force in the Steam Engine. 5th edition, revised and enlarged. 8vo. London, 1894. $3.00

Pray. Twenty Years with the Indicator. Being a Practical Text-book for the Engineer or the Student, with no Complex Formulæ. With many illustrations and rules as to the best way to run any Steam Engine to get the most economical results. How to Adjust Valves and Valve Motions Correctly. Full directions for working out Horse-Power, the Amount of Steam or Water per Horse-Power, Economy and Fuel. Extended directions for Attaching the Indicator, what Motions to use and those not to use. Full directions for Computation of Power by Planimeter and other methods, with many tables and hints. 8vo. New York, 1896. $2.50

INJECTORS.

Kneass. Practice and Theory of the Injector. 8vo. New York, 1898. $1.50

Nissenson. Practical Treatise on Injectors as Feeders of Steam Boilers. Illustrated. 8vo, paper. New York, 1890. $0.50

Pochet. Steam Injectors: Their Theory and Use. 16mo, boards. New York, 1890. $0.50

INSTRUCTIONS TO ENGINEERS, FIREMEN, AND BOILER ATTENDANTS.

Bale. A Hand-Book for Steam Users, being Rules for Engine Drivers and Boiler Attendants, with Notes on Steam Engine and Boiler Management and Steam Boiler Explosions. 12mo. London, 1890. $0.80

Edwards. 900 Examination Questions and Answers for Engineers and Firemen (Stationary and Marine), who desire to obtain a U. S. Government or State License. A new, revised, and enlarged edition. 32mo, mor. Philadelphia, 1901. $1.50

Grimshaw. Steam Engine Catechism. A Series of Direct Practical Answers to Direct Practical Questions. Mainly intended for Young Engineers. 18mo. New York, 1897. $2.00

Grimshaw. The Engine Runner's Catechism. Telling how to Erect, Adjust, and Run the principal Steam Engines in use in the United States. Illustrated. 18mo. New York, 1898. $2.00

Hawkins. Maxims and Instructions for the Boiler Room. Useful to Engineers, Firemen, and Mechanics, relating to Steam Generators, Pumps, Appliances, Steam Heating, Practical Plumbing, etc. 184 illustrations. 8vo. New York, 1901. $2.00

——**Aids to Engineers' Examinations.** Prepared for Applicants of all Grades with Questions and Answers. A Summary of the Principles and Practice of Steam Engineering. 12mo, leather, gilt edge. New York, 1901. $2.00

Reynolds. The Engineman's Pocket Companion and Practical Educator for Engineman, Boiler Attendants, and Mechanics. Illustrated. 16mo, 4th edition. London, 1900. $1.40

Roper. Instructions and Suggestions for Engineers and Firemen who wish to Procure a License, Certificate, or Permit to take charge of any class of Steam Engines or Boilers, Stationary, Locomotive, and Marine. 18mo, mor. Philadelphia, 1894. $2.00

Rose. Key to Engines and Engine-running. A Practical Treatise upon the Management of Steam Engines and Boilers for the use of those who desire to pass an examination to take charge of an engine or boiler. 12mo, cloth. New York, 1899. $2.50

—— **Questions and Answers for Engineers.** This little book contains all the questions that Engineers will be asked when undergoing an examination for the purpose of procuring licenses, and they are so plain that any Engineer or Fireman of ordinary intelligence may commit them to memory in a short time. 6th edition. 18mo, mor. Philadelphia. $2.00

Stephenson. Illustrated Practical Test Examination and Ready Reference Book for Stationary, Locomotive, and Marine Engineers, Firemen, Electricians, and Machinists, to procure Steam Engineer's license. 16mo, Chicago, 1892. $1.00

Stromberg. Steam User's Guide and Instructor. Plain and Correct Explanations in regard to Engines, Pumps, Dynamos, and Electricity. Practically, so that Engineers, Machinists, Firemen, and Electricians of Limited Education can understand and become expert practical engineers. 16mo. St. Louis, 15th edition. 1900. $2.50

Watson. How to Run Engines and Boilers. Practical Instruction for Young Engineers and Steam Users. 2d edition. Illustrated. 16mo. New York, 1896. $1.00

Zwicker. Practical Instructor in questions and answers for Machinists, Firemen, Electricians, and Steam Engineers. 24mo. St. Louis, Mo., 1898. $0.75

LOCOMOTIVE ENGINEERING.

Grimshaw. Locomotive Catechism. Containing nearly 1,300 Questions and Answers Concerning Designing and Constructing, Repairing and Running Various Kinds of Locomotive Engines. Intended as Examination Questions and to Post and Remind the Engine Runner, Fireman, or Learner. 176 illustrations. 12mo. New York, 1898. $2.00

Hill. Progressive Examinations of Locomotive Engineers and Firemen. 16mo. Terre Haute, Ind., 1899. $0.50

Meyer. Modern Locomotive Construction. 1,030 illustrations. 4to. New York, 1899. $10.00

Phelan. Air Brake Practice, being a description of the construction, objects sought, and results obtained, by the Westinghouse automatic air brake, as well as complete directions for operating it under the many diverse conditions in daily practice. 3 large folding plates. 12mo. New York. 4th edition. $1.00

Reagan. Locomotive Mechanism and Engineering. 12mo, with 145 illustrations. New York, 1898. $2.00

Reynolds. Locomotive Engine Driving. A Practical Manual for Engineers in charge of Locomotive Engines. 8th edition, enlarged. 12mo. London, 1892. $1.40

—— **The Model Locomotive Engineer,** Fireman, and Engine Boy: Comprising a Historical Notice of the Pioneer Locomotive Engines and their Inventors. 12mo. London, 1895. $1.80

—— **Continuous Railway Brakes.** A Practical Treatise on the several Systems in use in the United Kingdom; their Construction and Performance. Numerous illustrations and tables. 8vo. London, 1882. $3.60

—— **Engine Driving Life:** Stirring Adventures and Incidents in the Lives of Locomotive Engine Drivers. 2d edition, with additional chapters. 12mo. London, 1894. $0.80

Rogers. Pocket Primer or Air Brake Instruction. Stiff paper cover. $0.50

Roper. Hand-Book of the Locomotive; including the construction of engines and boilers and running of locomotives. 15th edition, revised. 12mo, mor. tucks. Philadelphia, 1897. $2.50

Sinclair. Locomotive-Engine Running and Management. A Practical Treatise on Locomotive Engines, showing their performance in running different kinds of trains with economy and Despatch. Also, directions regarding the care, management, and repairs of Locomotives and all their connections. Illustrated by numerous engravings. 21st edition, revised. 12mo. New York, 1901. $2.00

Stretton. The Locomotive Engine and its Development. A Popular Treatise on the Gradual Improvements made in Railway Engines between the years 1803 and 1892. Illustrated. 12mo. 5th edition. London, 1896. $1.50

Synnestvedt. Diseases of the Air Brake System. Their Causes, Symptoms, and Cure. Illustrated. 12mo. 1894. $1.00

Woods. Compound Locomotives. 2d edition, revised and enlarged by D. L. Barnes. 8vo. Illustrated. Chicago, 1894. $3.00

MACHINE TOOLS AND APPLIANCES.

Harrison. The Mechanic's Tool Book, with Practical Rules and Suggestions for Machinists, Iron Workers, and others. 12mo. New York, 1882. $1.50

Hasluck. The Mechanics' Work-shop Handy Book. A Practical Manual on Mechanical Manipulation. Embracing Information on Various Handicraf. Processes, with Useful Notes and Miscellaneous Memoranda. 12mo. London, 1895. $0.50

Knight. Mechanician. A Treatise on the Construction and Manipulation of Tools, for the Use and Instruction of Young Engineers and Scientific Amateurs. 4th edition. 4to. London, 1888. $7.25

Lukin. Young Mechanic. Containing directions for the use of all kinds of Tools and for construction of Steam Engines and Mechanical Models, including the Art of Turning in Wood and Metal. Illustrated. 12mo. New York. $1.75

Rose. Complete Practical Machinist. Embracing Lathe Work, Vise Work, Drills and Drilling, Taps and Dies, Hardening and Tempering, the Making and Use of Tools, Tool Grinding, Marking out Work, etc. Illustrated by 356 engravings. 19th edition, greatly enlarged. 12mo. Philadelphia, 1901. $2.50

Shelley. Work-shop Appliances. Including descriptions of some of the Gauging and Measuring Instruments, Hand Cutting Tools, Lathes, Drilling, Planing, and other Machine Tools used by Engineers. 10th edition, with an additional chapter on Milling, by R. R. Lister. Illustrated. 12mo. London, 1897. $1.50

Smith. Cutting Tools worked by Hand and Machine. 14 plates and 51 illustrations. 2d edition. 12mo. London, 1884. $1.50

Usher. Modern Machinist. A Practical Treatise on Modern Machine Shop Methods, describing in a comprehensive manner the most Approved Methods, Processes, and Appliances Employed in Present Practice, etc. 257 illustrations. 12mo. New York, 1895. $2.50

Watson. Modern Practice of American Machinists and Engineers. 12mo. Illustrated. Philadelphia, 1897. $2.50

MECHANICAL DRAWING AND MACHINE DESIGN.

Andre. Draughtsman's Hand-Book of Plan and Map Drawing; including Instructions for the preparation of Engineering, Architectural and Mechanical Drawings, with numerous illustrations, and colored examples. 8vo. London, 1891. $3.75

Appleton's Cyclopædia of Technical Drawing. Embracing the Principles of construction as applied to Practical Design. With numerous illustrations of Topographical, Mechanical, Engineering, Architectural, Perspective, and Free-hand Drawing. 8vo, leather. New York, 1896. $9.00

Barber. Engineers' Sketch Book of Mechanical Movements, Devices, Appliances, Contrivances, Details employed in the Design and Construction of Machinery for every Purpose. Collected from numerous sources and from actual work. Classified and arranged for reference. Nearly 2000 Illustrations. 4th edition. 8vo. London, 1902. $4.00

Building and Machine Draughtsman. A Practical Guide to the Projection and Delineation of Subjects met with in the practice of the engineer, machinist, and building constructor, etc.; by practical draughtsmen. 12mo. London, 1891. $2.00

Burns. Illustrated Architectural Engineering and Mechanical Drawing Book. For the use of Schools, Students, and Artisans. 10th edition, revised and corrected, with additional sections on important departments of the art. 8vo. 284 illustrations. New York, 1893. $1,00

Cathcart, Prof. Wm. L. Machine Design: Fastenings, with Tables, Diagrams and Engravings. 8vo, cloth. Illustrated. *In Press.*

Davidson. Drawing for Machinists and Engineers. Comprising a complete course of Drawing adapted to the requirements of Millwrights and Engineers; also, course of practical instruction in the coloring of mechanical drawings. 4th edition. 16mo. London. $1.75

Donaldson. Drawing and Rough Sketching for Marine Engineers, with Proportions, Instructions, Explanations, and Examples; also How to Design Engines, Boilers, Propellers, Paddle Wheels, Shafts, Rods, Valves, etc. 6th edition. Illustrated. London, 1899. $3.00

Faunce. Mechanical Drawing, prepared for the use of the students of the Mass. Institute of Technology. 2d edition, revised and enlarged. Illustrated and 8 plates. 12mo. Boston, 1900. $1.25

Fox, Wm., and C. W. Thomas, M. E. A Practical Course in Mechanical Drawing. 2d edition, revised. 12mo, cloth, with plates. $1.25

Halliday. First Course in Mechanical Drawing (Tracing). Folio, paper. London, 1889. $0.75

——— **Mechanical Graphics.** A Second Course in Mechanical Drawing, with Preface by Professor Perry. 8vo. London, 1889. $2.00

Hulme. Mathematical Drawing Instruments and How to Use Them. 4th edition. 12mo. New York, 1890. $1.50

Klein. Elements of Machine Design. Notes and Plates, 8vo. Bethlehem, Pa., 1892. $6.00

Low and Bevis. Manual of Machine Drawing and Design. 3d edition, 753 illustrations. 8vo. London, 1901 $2.50

MacCord. Practical Hints for Draughtsmen. Illustrated with 68 diagrams and full page plates. 3d edition, 4to. New York, 1890. $2.50

—— **Kinematics,** or Practical Mechanics. A Treatise on the Transmission and Modification of Motion and the Construction of Mechanical Movements. For the use of Draughtsmen, Machinists, and Students of Mechanical Engineering, in which the laws governing the motions and various parts of Mechanics, as affected by their forms and modes of connection, are deduced by simple geometrical reasoning, and their application is illustrated by accurately constructed diagrams of the different mechanical combinations discussed. 4th edition. 8vo. New York, 1899. $5.00

Minifie. Mechanical Drawing. A Text-Book of Geometrical Drawing, for the use of Mechanics and Schools, in which the Definitions and Rules of Geometry are familiarly explained: the Practical Problems are arranged from the most simple to the more complex, and in their description technicalities are avoided as much as possible. With illustrations for Drawing Plans, Sections, and Elevations of Buildings and Machinery; an Introduction to Isometrical Drawing, and an Essay on Linear Perspective and Shadows. Illustrated by over 200 diagrams, engraved on steel. With an Appendix on the Theory and Application of Colors. 8vo. New York, 1893. $4.00

—— **Geometrical Drawing.** Abridged from the octavo edition, for the use of Schools. Illustrated with 48 steel plates. 9th edition. Revised and enlarged. 12mo. New York, 1890. $2.00

Palmer. Mechanical Drawing, Projection Drawing, Geometric and Oblique Drawing, Working Drawings. A Condensed Text for Class Room use. 8vo. Columbus, O. 1894. $1.00

Reinhardt, Chas. W. Lettering for Draftsmen, Engineers and Students. A Practical System of Free-hand Lettering for Working Drawings. 11th thousand. Oblong, boards. $1.00

——— **The Technic of Mechanical Drafting.** A Practical Guide to neat, correct and legible Drawing, containing many Illustrations, Diagrams and full-page Plates. 4to, cloth. Illustrated. $1.00

Ripper. Machine Drawing and Design for Technical Schools and Engineer Students. Being a complete course of Instruction in Engineering Drawing, with Notes and Exercises on the Application of Principles to Engine and Machine Design, and on the Preparation of Finished Colored Drawings. Illustrated by 52 plates and numerous explanatory drawings. 8vo. London, 1897. $6.00

Roberts. Drawing and Designing for Marine Engineers. 21 large folding plates and many other illustrations throughout the text. 8vo. London, 1898. $3.00

Rose. Mechanical Drawing Self-Taught. Comprising Instructions in the Selection and Preparation of Drawing Instruments, Elementary Instruction in Practical Mechanical Drawing, together with Examples in Simple Geometry and Elementary Mechanism, including Screw Threads, Gear Wheels, Mechanical Motions, Engines and Boilers. Illustrated by 330 engravings. 4th edition, revised. 8vo. Philadelphia, 1902. $4.00

Shaw. Mechanical Integrators. Including the various Forms of Planimeters. 18mo, boards. Illustrated. New York, 1886. $0.50

Smith. Graphics, or the Art of Calculation by Drawing Lines, applied especially to Mechanical Engineering. Part I. Text, with Separate Atlas of Plates — Arithmetic, Algebra, Trigonometry, Vector and Lecor Addition, Machine Kinematics, and Statics of Flat and Solid Structures. 8vo. London, 1888. $5.00

Stanley. Descriptive Treatise on Mathematical Drawing Instruments, their Construction, Uses, Qualities, Selection, Preservation, and Suggestions for Improvements, with Hints upon Drawing and Coloring. 5th edition. 12mo. London, 1878. $2.00

Tomkins. Principles of Machine Construction; being an application of Geometrical Drawing for the Representation of Machinery. Text 12mo, Plates 4to. New York. $3.50

Unwin. Elements of Machine Design. Part I. General Principles, Fastenings, and Transmissive Machinery. 16th edition. 12mo. London, 1898. $2.00

—— Part II. Chiefly on Engine Details. 12mo. 13th edition, revised and enlarged. London, 1901. $1.50

Warren. Elements of Machine Construction and Drawing: or, Machine Drawing, with some elements of descriptive and rational kinematics. 2 vols. Text and plates. 8vo. New York. $7.50

MECHANICAL ENGINEERS' HAND-BOOKS.

Adams. Hand-Book for Mechanical Engineers. 2d edition. Revised and enlarged. 12mo. London, 1897. $2.50

Appleton's Cyclopædia of Applied Mechanics: a Dictionary of Mechanical Engineering and the Mechanical Arts. Edited by Park Benjamin. Nearly 7,000 illustrations. Revised and improved edition. 2 vols. 8vo, leather. New York, 1896. $15.00

Bale. Steam and Machinery Management: A Guide to the Arrangement and Economical Management of Machinery, with Hints on Construction and Selection. Illustrated. 2d edition. 12mo. London, 1890. (Weale's Series.) $1.00

Benjamin. Wrinkles and Recipes. Compiled from the *Scientific American*. A collection of Practical Suggestions, Processes, and Directions, for the Mechanic, Engineer, Farmer, and Housekeeper. With a Color Tempering Scale and numerous Wood Engravings. 5th edition. 12mo. New York, 1901. $2.00

Byrne. Hand-Book for the Artisan, Mechanic, and Engineer. Comprising the Grinding and Sharpening of Cutting Tools, Abrasive Processes, Lapidary Work, Gem and Glass Engraving, Varnishing and Lackering Apparatus, Materials and Processes for Grinding and Polishing, etc. 8vo. Illustrated. Philadelphia, 1887. $5.00

Carpenter. Text-Book of Experimental Engineering. For Engineers and for Students in Engineering Laboratories. 249 illustrations. 5th revised edition. 8vo. New York, 1898. $6.00

Chordal. Extracts from Chordal's Letters. Comprising the choicest selections from the Series of Articles which have been appearing for the past two years in the columns of the *American Machinist*. With over 50 illustrations. 12mo. 9th edition. New York, 1901. $2.00

Clark. Manual of Rules, Tables and Data for Mechanical Engineers, based on the most recent investigations. With numerous Diagrams. 7th edition. 1,012 pages. London, 1897. $5.00
Half morocco. $7.50

——— **Mechanical Engineers' Pocket-Book** of Tables, Formulæ, Rules, and Data. A Handy-Book of Reference for Daily Use in Engineering Practice. 16mo, mor. 4th edition. London, 1899. $3.00

Dixon. The Machinists' and Steam Engineers' Practical Calculator. A Compilation of useful Rules and Problems, arithmetically solved, together with general information applicable to Shop Tools, Mill Gearing, Pulleys and Shafts, Steam Boilers and Engines. Embracing valuable Tables and Instructions in Screw Cutting, Valve and Link Motion. 3d edition. 16mo, mor., pocket form. New York, 1901. $1.25

Engineering Estimates, Costs, and Accounts. A Guide to Commercial Engineering. With numerous Examples of Estimates and Costs of Millwright Work, Miscellaneous Productions, Steam Engines and Steam Boilers, and a Section on the Preparation of Costs Accounts. By a General Manager. 2d edition. 8vo. London, 1896. $4.80

General Machinist, Being a Practical Introduction to the Leading Departments of Mechanism and Machinery, the Communication of Motion or the Transmission of Force by Belt, Rope, Wire Rope, and Pulley Gearing — Toothed-Wheel and Frictional Gearing; together with the details of the component and essential parts of mechanism — Shafts, Pedestals, Hanger, Clutches, etc., and of the methods of fitting up Machines, Screw Bolts, Riveting, etc. By various practical writers and machinists. 75 illustrations and 4 folding plates. 8vo. London, 1891. $2.00

Grimshaw. Hints to Power Users. Plain, Practical Pointers, free from high Science, and intended for the man who pays the bills. 12mo. New York, 1891. $1.00

Hasluck. Mechanic's Workshop Handy-Book. A Practical Manual on Mechanical Manipulation. Embracing Information on Various Handicraft Processes, with Useful Notes, and Miscellaneous Memoranda. 12mo. London, 1888. $0.50

Haswell. Engineers' and Mechanics' Pocket Book, Containing Weights and Measures, Rules of Arithmetic, Weights and Materials, Latitude and Longitude, Cables and Anchors, Specific Gravities, Squares, Cubes, and Roots, etc.; Mensuration of Surfaces and Solids, Trigonometry, Mechanics, Friction, Aerostatics, Hydraulics and Hydrodynamics, Dynamics, Gravitation, Animal Strength, Windmills, Strength of Materials, Limes, Mortars, Cements, etc.; Wheels, Heat, Water, Gunnery, Sewers, Combustion, Steam and the Steam Engine, Construction of Vessels, Miscellaneous Illustrations, Dimensions of Steamers, Mills, etc.; Orthography of Technical Words and Terms, etc. 67th edition. Revised and enlarged. 12mo, mor. tuck. New York, 1902. $4.00

Hawkins. Hand-Book of Calculations, for Engineers and Firemen; relating to the Steam Engine, the Steam Boiler, Pumps, Shafting, etc. Illustrated. 8vo. New York, 1902. $2.00

Hutton. Works Manager's Hand-Book of Modern Rules, Tables, and Data for Civil and Mechanical Engineers, Millwrights, and Boiler Makers, Tool Makers, Machinists, and Metal Workers, Iron and Brass Founders, etc. 5th edition, revised, with additions. 8vo, half-bound. London, 1895. $6.00

Hutton. Practical Engineer's Hand-Book. Comprising a Treatise on Modern Engines and Boilers, Marine, Locomotive, and Stationary, and containing a large Collection of Rules and Practical Data Relating to Recent Practice in Designing and Constructing all kinds of Engines, Boilers, and other Engineering Work. 5th edition, carefully revised, with additions. 370 illustrations. 8vo. London, 1896. $7.00

Kent. Mechanical Engineers' Pocket-Book. A Reference Book of Rules, Tables, Data, and Formulæ, for the Use of Engineers, Mechanics, and Students. 1,087 pages. 5th edition. 12mo. New York, 1902. $5.00

Knight. American Mechanical Dictionary. A Descriptive Word Book of Tools, Instruments, Chemical and Mechanical Processes; Civil, Mechanical, Railroad, Hydraulic, and Military Engineering. A History of Inventions. General Technological Vocabulary, and Digest of Mechanical Appliances in Science and the Industrial and Fine Arts. 3 vols. Illustrated, 8vo. Boston, 1884. $24.00

Supplement to the above, $9.00

The 4 vols., complete, $27.50

Lockwood's Dictionary of Terms used in the Practice of Mechanical Engineering. Embracing those current in the Drawing Office, Pattern Shop, Foundry, Fitting, Turning, Smiths' and Boiler Shops, etc., comprising upwards of 6,000 definitions. Edited by a Foreman Pattern Maker. 3d edition. 12mo. London, 1902. $3.00

Molesworth. Pocket-Book of Useful Formulæ and Memoranda for Civil and Mechanical Engineers. 24th edition, revised and enlarged. Pocket-book form. London, 1901. $2.00

Moore. Universal Assistant and Complete Mechanic: Containing over One Million Industrial Facts, Calculations, Receipts, Processes, Trade Secrets, Rules, Business Forms, Legal Items, etc. Illustrated. 12mo. New York. $2.50

Rankine. Useful Rules and Tables relating to Mensuration, Engineering Structures, and Machines. 7th edition, thoroughly revised by W. J. Millar. With Electrical Engineering Tables, Tests, and Formulæ for the use of Engineers, by Prof. A. Jamieson. 12mo. London, 1889. $4.00

Roper. Engineers' Handy-Book. Containing a full explanation of the Steam Engine Indicator, and the Use and Advantage to Engineers and Steam Users. With Formulæ for estimating the Power of all Classes

of Steam Engines; also Facts, Figures, Questions, and Tables, for Engineers who wish to qualify themselves for the United States Navy, the Revenue Service, the Mercantile Marine, or to take charge of the better class of stationary Steam Engines. Illustrated. 15th edition. 16mo, mor. tucks. Philadelphia, 1901. $3.50

Scribner. Engineers' and Mechanics' Companion. Comprising United States Weights and Measures, Mensuration of Superfices and Solids; Tables of Squares and Cubes; Square and Cube Roots; Circumference and Areas of Circles; the Mechanical Powers; Centres of Gravity; Gravitation of Bodies; Pendulums; Specific Gravity of Bodies; Strength, Weight, and Crush of Materials; Water-wheels, Hydrostatics, Hydraulics, Statics, Centres of Percussion and Gyration; Friction Heat; Tables of the Weight of Metals, Scantling, etc.; Steam and Steam Engine. 21st edition, revised. 16mo, full mor. New York, 1902. $1.50

Spons' Tables and Memoranda for Engineers, and convenient reference for the pocket. 12th edition. 64mo, roan, gilt edges. London, 1901. In cloth case. $0.50

——— **Mechanics' Own Book.** A Manual for Handicraftsmen and Amateurs. Complete in one large vol., 8vo, containing 700 pages and 1,420 Illustrations. 6th edition. London, 1901. $2.50

——— **Dictionary of Engineering.** Civil, Mechanical, Military, and Naval, with Technical Terms in French, German, Italian, and Spanish. 8 vols. 8vo, cl. London, 1874. Each, $5.00

——— Supplement to above. 3 vols., cl. London, 1881. Each, $5.00

Templeton. Practical Mechanics' Workshop Companion. Completing a great variety of the most useful Rules and Formulæ in Mechanical Science, with numerous Tables of Practical Data and Calculated Results for Facilitating Mechanical Operations. 18th edition, revised, modernized, and considerably enlarged, by Walter S. Hutton. 16mo, leather. London, 1902. $2.00

——— **Engineers', Millwrights', and Mechanics' Pocket Companion.** Comprising Decimal Arithmetic, Tables of Square and Cube Roots, Practical Geometry, Mensuration, Strength of Materials, Mechanical Powers, Water Wheels, Pumps and Pumping Engines, Steam Engines, Tables of Specific Gravity, etc. Revised, corrected, and enlarged from the 8th English edition, and adapted to American Practice, with the addition of much new matter. Illustrated by J. W. Adams. 12mo, mor. tucks. New York, 1893. $2.00

Van Cleve. English and American Mechanic. An every-day Hand-Book for the Workshop and the Factory. Containing Several Thousand Receipts, Rules, and Tables indispensable to the Mechanic, the Artisan, and the Manufacturer. A new, revised, enlarged, and improved edition. Edited by Emory Edwards, M.E. 12mo. Philadelphia, 1893. $2.00

MECHANICS (ELEMENTARY AND APPLIED).

Church. Notes and Examples in Mechanics; with an Appendix on the Graphical Statics of Mechanism. 128 illustrations and 6 plates. 8vo. 2d edition, New York, 1900. $2.00

Cotterill. Applied Mechanics, an Elementary General Introduction to the Theory of Structures and Machines. Illustrated. 3d edition. 8vo. London, 1895. $5.00

—— **and Slade. Lessons in Applied Mechanics.** 12mo. London, 1894. *Net* $1.25

Dana. A Text-Book of Elementary Mechanics for the use of Colleges and Schools. 12th edition. 12mo. New York, 1901. $1.50

DuBois. Elementary Principles of Mechanics. Designed as a Text-Book for technical schools. 3 vols. 8vo. New York.

Vol. I. Kinematics. $3.50
Vol. II. Statics. $4.00
Vol. III. Kinetics. $3.50

Garnett. Treatise on Elementary Dynamics. For the use of Colleges and Schools. 5th edition. 8vo. London, 1889. *Net* $1.50

Geldard. Statics and Dynamics. Illus. 12mo. London, 1893. $1.50

Goodeve. Principles of Mechanics. New edition, rewritten and enlarged. 12mo. London, 1889. $2.50

—— **Manual of Mechanics.** An Elementary Text-Book for Students of Applied Mechanics. Illustrated. 12mo. London, 1881. $0.80

Hancock. Text-Book of Mechanics and Hydrostatics. With over 500 diagrams. 8vo. New York, 1894. $1.75

Hughes. Condensed Mechanics: a selection of Formulæ, Rules, Tables, and Data for the Use of Engineering Students, Science Classes, etc., in accordance with the requirements of the Science and Art Department. 12mo. London, 1891. $1.00

Jamieson. Elementary Manual of Applied Mechanics. Specially arranged for the use of First Year Science and Art, City and Guilds of London Institute, and other Elementary Engineering Students. 4th edition, revised and enlarged. 12mo. London, 1900. $1.25

Kennedy. Mechanics of Machinery. With numerous illustrations. 12mo. London, 1886. $3.50

—— **Kinematics of Machinery;** or, The Elements of Mechanism. 16mo, boards. New York, 1881. $0.50

Nystrom. New Treatise on Elements of Mechanics. 8vo. Philadelphia, 1875. $2.00

Perry. Applied Mechanics. Illustrated. 12mo. London, 1901. $2.50

Practical Mechanics. Being the Fourth Volume of "Amateur Work Illustrated." Plates and illustrations. 4to. London. $3.00

Rankine. Applied Mechanics, comprising Principles of Statics, Cinematics, and Dynamics, and Theory of Structures, Mechanism, and Machines. 12mo. 16th edition, thoroughly revised, by W. J. Millard. London, 1901. $5.00

——— **and Bamber. Mechanical Text-Book;** or, Introduction to the Study of Mechanics and Engineering. With numerous Diagrams. 4th edition, revised. 8vo. London, 1890. $3.50

Stahl and Woods. Elementary Mechanism. A Text-Book for Students of Mechanical Engineering. 11th edition, revised and enlarged. Illustrated. 12mo. New York, 1901. $2.00

Weisbach. Theoretical Mechanics, with an Introduction to the Calculus. Translated from the 4th German edition by E. B. Coxe. 9th edition, revised. 8vo, cloth. New York, 1899. $6.00
Sheep. $7.00
Vol. II., Part 1. Hydraulics and Hydraulic Motors. $5.00
Vol. II., Part 2. Heat, Steam, and Steam Engines. $5.00
Vol. III., Part 1. Kinematics and Machinery of Transmission. $5.00
Vol. III., Part 2. Machinery of Transmission and Governors. $5.00

Wood. Elements of Analytical Mechanics. With numerous examples and illustrations. For use in Scientific Schools and Colleges. 7th edition, revised and enlarged, comprising Mechanics of Solids and Mechanics of Fluids, of which Mechanics of Thirds is entirely new. 8vo. New York, 1900. $3.00

—— **Principles of Elementary Mechanics.** Fully illustrated. 9th edition. 12mo. New York, 1894. $1.25

Wright. Text-Book of Mechanics. With numerous examples. 3d edition. 12mo. New York. $2.50

MISCELLANEOUS.

Amateur Mechanic's Workshop. A Treatise containing plain and concise directions for the manipulation of Wood and Metals, including Casting, Forging, Brazing, Soldering, and Carpentry. By the author of "The Lathe and its Uses." 7th ed. Illustrated. 8vo. London, 1888. $3,00

Hiscox, Gardner D. Compressed Air; Its Production, Uses, and Applications. Comprising the Physical Properties of Air from a Vacuum to its Liquid State, its Thermodynamics, Compression, Transmission, and Uses as a Motive Power. With 40 Air Tables and 545 Illustrations. 8vo, cloth. New York, 1901. $5.00
Half morocco. $6.50

Plympton, Prof. Geo. W. How to become an Engineer; or, the Theoretical and Practical Training necessary in fitting for the Duties of the Civil Engineer. (Van Nostrand's Science Series). $0.50

STEAM AND STEAM ENGINES.

Alexander. Model Engine Construction. With Practical Instructions to Artificers and Amateurs. Containing numerous illustrations and twenty-one Working Drawings, from Original Drawings by the Author. 12mo. London, 1895. $3.00

Baker. Treatise on the Mathematical Theory of the Steam Engine. With Rules at length and Examples worked out, for the use of practical men, with numerous diagrams. 8th edition. London, 1890. $0.60

Bale. How to Manage a Steam Engine; a Handbook for all who use Steam. Illustrated, with examples of different Types of Engines and Boilers; with Hints on their Construction, Working, Fixing, Economy of Fuel, etc. 7th edition. 12mo. London, 1890. $0.80

Barrus, Geo. H., S. B. Engine Tests; embracing the results of over 100 Feed-Water Tests and other investigations of various kinds of Steam Engines, conducted by the author. Illustrated. 8vo, cloth. New York, 1900. $4.00

Bourne. Catechism of the Steam Engine in its various Applications to Mines, Mills, etc. New edition, enlarged. Illustrated. 12mo. New York, 1897. $2.00

—— **Hand-Book of the Steam Engine,** containing all the Rules required for the right Construction and Management of Engines of every Class with the easy Arithmetical Solution of those Rules. Illustrated. 12mo New York. 1892. $1.75

Burn. Steam Engine, its History and Mechanism. 3d edition. 8vo. Illustrated. London, 1857. $1.00

Clark. Steam and the Steam Engine, Stationary and Portable. (Being an Extension of the Elementary Treatise on the Steam Engine, of Mr. John Sewell.) 4th edition. London, 1892. $1.40

—— **The Steam Engine.** A Treatise on Steam Engines and Boilers; comprising the Principles and Practice of the Combustion of Fuel, the Economical Generation of Steam, the Construction of Steam Boilers, and the principles, construction, and performance of Steam Engines, Stationary, Portable, Locomotive, and Marine, exemplified in Engines and Boilers of recent date. Illustrated by above 1,300 figures in the text, and a series of folding plates drawn to scale. 2 vols. 8vo. London, 1895. $15.00

Colyer. Treatise on Modern Steam Engines and Boilers, including Land, Locomotive, and Marine Engines and Boilers. For the use of Students. With 46 plates. 4to. London, 1886. $5.00

Cotterill. Steam Engine considered as a Thermodynamic Machine. A Treatise on the Thermodynamic Efficiency of Steam Engines. Illustrated by tables, diagrams, and examples from practice. 3d edition, revised and enlarged. 8vo. London. 1896. *net* $4.50

Diesel. Theory and Construction of a Rational Heat Motor. Translated from the German by Bryan Donkin. With eleven figures in the text and three plates. 8vo. London, 1894. $2.50

Edwards. American Steam Engineer, Theoretical and Practical. With Examples of the latest and most approved American Practice on the Design and Construction of Steam Engines and Boilers of every description. For the use of Engineers, Machinists, Boiler Makers, etc. Illustrated by 77 engravings. 12mo. Philadelphia, 1897. $2.50.

—— **Practical Steam Engineers' Guide** in the Design, Construction, and Management of American Stationary, Portable, and Steam Fire Engines, Steam Pumps, Boilers, Injectors, Governors, Indicators, Pistons, and Rings, Safety Valves and Steam Gauges. For the use of Engineers, Firemen, and Steam Users. Illustrated. 3d edition, revised and corrected. 12mo. Philadelphia, 1900. $2.50

Evers. Steam and other Prime Movers. A Text-Book both Theoretical and Practical. Illustrated. 12mo. London, 1890. $1.50

—— **Steam and the Steam Engine;** Land, Marine, and Locomotive Illustrated. 12mo. New York. $1.00

Ewing. Steam Engine and other Heating Engines. Illustrated. 8vo. Cambridge, 1899. $3.75

Goodeve. Text-Book on the Steam Engine. With a Supplement on Gas Engines and on Heat Engines. 13th edition, enlarged. 12mo. 143 illustrations. London, 1896. $2.00

Gould. Arithmetic of the Steam Engine. 12mo. N. Y. 1898. $1.00

Grimshaw. Steam Engine Catechism. A series of direct practical answers to direct practical questions, mainly intended for young engineers and for examination questions. 11th edition, enlarged and improved. 18mo. New York, 1897. $2.00

Haeder. Hand-Book on the Steam Engine with especial Reference to Small and Medium sized Engines. For the Use of Engine Makers, Mechanical Draughtsmen, Engineering Students, and Users of Steam Power. 1,100 illustrations. 12mo. London, 1896. $3.00

Henthorn. Corliss Engine and its Management. Edited by E. P. Watson. 3d edition, enlarged with an appendix, by Emil Herter. Illustrated. 18mo. New York, 1897. $1.00

Holmes. Steam Engine. 212 illustrations. 10th edition. 12mo. London, 1900. $2.00

This is a complete practical and theoretical treatise on the steam-engine, written in very clear and beautiful style, rendering the more abstruse principles of the subject as plain and simple as it is probably possible to make them. It is one of the best, if not the best, combinations of theoretical investigation and practical applications in the whole literature of the subject, and forms an admirable companion to Ripper's smaller and more exclusively practical treatise.

Jamieson. Text-Book of Steam and Steam Engines. 13th edition, with numerous Diagrams, four folding Plates, and Examination Questions. 12mo. London, 1901. $3.00

——— **Elementary Manual on Steam and the Steam Engine.** With numerous Diagrams, Arithmetical Examples, and Examination Questions. 8th edition. 12mo. London, 1900. $1.40

Lardner. Treatise on the Steam Engine, for the Use of Beginners. 16th edition. Illustrated. London, 1893. $0.60

Le Van. Steam Engine and the Indicator; their Origin and Progressive Development, including the most recent examples of Steam and Gas Motors, together with the Indicator, its Principles, its Utility, and its Application. Illustrated by 205 Engravings, chiefly of Indicator Cards. 8vo. Philadelphia, 1900. $4.00

Mallet. Compound Engines. 16mo, boards. New York, 1884. $0.50

Marks. Relative Proportions of the Steam Engine. Illustrated. 3d edition. 12mo. Philadelphia, 1896. $3.00

Peabody. Table of the Properties of Saturated Steam and other Vapors. 6th edition. 8vo. New York, 1901. $1.00

Pray. Steam Tables and Engine Constants. For facilitating all calculations upon Indicator Diagrams or Various Problems connected with the operation of the Steam Engine, from reliable data and with precision compiled from Regnault, Rankine, and Dixon directly, making use of the exact records. 8vo. New York, 1894. $2.00

Rankine. Manual of the Steam Engine and other Prime Movers, with numerous tables and illustrations. 12mo. 14th edition. London, 1897. $5.00

Rigg. Practical Treatise on the Steam Engine, containing Plans and Arrangements of Details for Fixed Steam Engines, with Essays on the Principles involved in Design and Construction. Copiously illustrated with woodcuts and 96 plates. 4to. 2d edition. New York, 1894. $10.00

Ripper. Steam. Illustrated. 12mo. London, 1889. $1.00

This work is based upon a course of lectures given to an evening class of young mechanical engineers on steam, steam-engines, and boilers. It is remarkably clear, concise, and practical; no superfluous matter is introduced, and every page goes directly to the point. It is the best book for beginners, and also for those who wish to have a manual embracing the practical features of the subjects in small compass.

Roper. Hand-Book of Modern Steam Fire Engines; including the running, care, and management of Steam Fire Engines and Fire Pumps. 2d edition, revised and corrected by H. L. Stellwagen. Illustrated. 12mo, mor. tucks. Philadelphia, 1897. $3.50

—— **Hand-Book of Land and Marine Engines,** including the Modelling, Construction, Running, and Management of Land and Marine Engines and Boilers. 9th edition, revised, enlarged, and improved. 12mo, mor. tucks. Philadelphia, 1897. $3.50

—— **Catechism** of High Pressure or Non-Condensing Steam Engines, including the Modelling, Constructing, Running, and Management of Steam Engines and Steam Boilers. 20th edition, revised and enlarged. Illustrated. 12mo, mor. tucks. Philadelphia, 1897. $2.00

—— **Young Engineer's Own Book.** Containing an Explanation of the Principle and Theories on which the Steam Engine as a Prime Mover is based, with a description of different kinds of Steam Engines, Condensing and Non-Condensing, Marine, Stationary, Locomotive, Fire, Traction, and Portable. 106 illustrations. 3d edition, revised. 16 mo, mor. tucks. Philadelphia, 1897. $3.00

Rose. Modern Steam Engines. An Elementary Treatise upon the Steam Engine, written in Plain Language; for use in the Workshop as well as in the Drawing Office. Giving Full Explanations of the Con-

struction of Modern Steam Engines; including Diagrams showing their Actual Operation; together with Complete but Simple Explanation of the Operations of various kinds of Valves, Valve Motions, and Link Motions, etc., thereby enabling the ordinary engineer to clearly understand the Principles involved in their Construction and use, and to Plot out their movements upon the Drawing Board. New edition, revised and improved. 453 illustrations. 4to. Philadelphia, 1900. $6.00

—— **Key to Engines and Engine Running.** A Practical Treatise upon the Management of Steam Engines and Boilers, for the use of those who desire to pass an Examination to take Charge of an Engine or Boiler. With numerous Illustrations and Instructions upon Engineers' Calculations, Indicator Diagrams, Engine Adjustments, and other Valuable Information necessary for Engineers and Firemen. 12mo. N. Y. 1899. $2.50

Thurston. History of the Growth of the Steam Engine. 4th revised edition. Illustrated. 12mo. New York, 1897. $2.50

—— **Manual of the Steam Engine.** For Engineers and Technical Schools. Part I, Structure and Theory. Illustrated. 5th edition, revised. 8vo. New York, 1900. $6.00

Part II. Design, Construction, and Operation. Illustrated. 4th edition, revised. 8vo. New York, 1900. $6.00

Or in sets. $10.00

——— **Hand-Book of Engine and Boiler Trials,** and of the Indicator and Prony Brake, for Engineers and Technical Schools. Illustrated. 4th and revised edition. 8vo. New York, 1897. $5.00

——— **Stationary Steam Engines,** Simple and Compound, especially as adapted to Electric Lighting Purposes. 5th edition, revised, with additions. Illustrated, 12mo. New York, 1893. $2.50

Turnbull. Treatise on the Compound Engine. 2d edition, revised and enlarged by Prof. S. W, Robinson. 16mo, boards. New York, 1884. $0.50

Watson, E. P. Small Engines and Boilers. A manual of Concise and Specific Directions for the Construction of Small Steam Engines and Boilers of Modern Types from five Horse-power down to model sizes. Illustrated. 12mo, cloth. $1.25

Weisbach. Heat, Steam, and Steam Engine. Translated from the 4th edition of Vol. II. of Weisbach's Mechanics. Containing Notes giving practical examples of Stationary, Marine, and Locomotive Engines, showing American practice, by R. H. Buel. Numerous illustrations. 8vo. New York, 1891. $5.00

Whitham. Steam Engine Design. For the use of Mechanical Engineers, Students, and Draughtsmen. 3d edition, revised. With 210 illustrations. 8vo. New York, 1902. $5.00

TRANSMISSION OF POWER, BELTING, ETC.

Kerr, E. W., M.E. Power, and Power Transmission. A Series of Lectures delivered to Students, on the Elementary Principles of Engineering. Containing numerous full-page Diagrams, Figures and Tables. Illustrated. 8vo, cloth. New York, 1902. $2.00

Toothed Gearing. A Practical Hand-Book for Offices and Workshops. By a Foreman Pattern Maker. 184 illustrations. 12mo. London, 1892. $2.50

Unwin. On the Development and Transmission of Power from Central Stations. Being the Howard Lectures delivered at the Society of Arts in 1893. Illustrated. 8vo, New York, 1894. $3.50

VALVES AND VALVE GEARS.

394
O

Auchincloss. Practical Application of the Slide-Valve and Link-Motion to Stationary, Portable, Locomotive, and Marine Engines, with new and simple methods for proportioning the parts. Illustrated. 14th edition, revised and enlarged. 8vo. New York, 1901. $2.00

Bankson. Slide Valve Diagrams. A French Method of Obtaining Slide Valve Diagrams. 8 Plates. 16mo. New York, 1892. $0.50

Begtrup, Julius. Slide Valve and its Functions. With special reference to modern Practice in the United States. With many Diagrams. Illustrated. 8vo, cloth. $2.00

Buel. Safety Valves. 3d edition. 16mo, boards. New York, 1898. $0.50

Halsey. Slide Valve Gears; an explanation of the Action and Construction of plain and cut-off Slide Valves. *Analysis by the Bilgram Diagram.* 79 illustrations. 7th edition, revised and enlarged. 12mo. New York, 1901. $1.50

—— **Worm and Spiral Gearing.** With Illustrations and Diagrams. (Van Nostrand's Science Series, No. 116.) $0.50

Le Van. Safety Valves; Their History, Antecedents, Invention, and Calculation. 69 Illustrations. 12mo. New York, 1892. $2.00

MacCord. Treatise on the Movement of the Eccentric upon the Slide Valve, and explaining the Practical Process of Laying out the Movements, adapting the Valve for its various duties in the Steam Engine, for the Use of Engineers, Draughtsmen, Machinists, and Students of Valve Motion in general. 2d edition. 4to. Illustrated. New York, 1883. $2.50

Peabody. Valve Gears and Steam Engines. 33 Plates. 8vo. New York, 1900. $2.50

Rose. Slide Valve Practically Explained. Embracing Simple and Complete Practical Demonstrations of the Operations of each Element in a Slide-Valve Movement, and illustrating the effects of variations in their proportions, by examples carefully selected from the most recent and successful practice. Illustrated. 12mo. Philadelphia, 1895. $1.00

Spangler. Valve Gears. 2d edition, revised and enlarged. 8vo. New York, 1900. $2.50

Welch. Treatise on a Practical Method of Designing Slide Valve Gearing by Simple Geometrical Construction, based upon the principles enunciated in Euclid's Elements, and comprising the various forms of Plain Slide Valve and Expansion Gearing; together with Stephenson's, Gooch's, and Allen's Link Motions, as applied either to reversing or to variable expansion combinations. 12mo. London, 1875. $1.50

Zeuner. Treatise on Valve Gears, with Special consideration of the link motions of locomotive engines. 4th edition. Translated by Prof. J. F. Klein. 8vo. London, 1884. $5.00

HALLIDIE LIBRARY
OF THE
UNIVERSITY
OF
CALIFORNIA

UNIVERSITY OF CALIFORNIA LIBRARY
BERKELEY

THIS BOOK IS DUE ON THE LAST DATE STAMPED BELOW

Books not returned on time are subject to a fine of 50c per volume after the third day overdue, increasing to $1.00 per volume after the sixth day. Books not in demand may be renewed if application is made before expiration of loan period.

NOV 16 1932

DEC 22 1932

1932

RECD CIRC MAR 7 1986

MAR 20 1986

50m-7,'16

GENERAL LIBRARY - U.C. BERKELEY

B000956615

YC 1289